AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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Detail Refinement

COMPARED with equivalent models of eleven or twelve years ago, the contemporary low and medium priced vehicles evidence an altogether remarkable advance in refinement. There are, however, still many detail points in need of attention, some of which are the more surprising in that they are immediately apparent to anyone on first driving a car and so can hardly have escaped the attention of the development department. Further, many of these minor irritations are no new phenomena, but have been tolerated for very many years, in the same way that seating positions in many cars are still designed to suit some "standardized" human form having proportions with no biological counterpart.

Quite one of the most unpleasant features of some recent models is the operation of the throttle by means of a flexible cable. Of all controls least suited to such a medium, the throttle is surely at the head of the list. Indeed, it stands alone, since not only is it in itself an extremely light movement, but it is also, with the possible exception of the steering wheel, by far the most frequently used control, calling for constant fine variation of setting in modern traffic conditions.

Faulty ratios

Unfortunately this type of control often appears to be associated with a badly chosen ratio between pedal and throttle spindle, whereby the most sensitive part of the throttle opening is accomplished with an apparently trifling movement of the foot. A badly chosen ratio of this kind is alone unpleasant enough, but in combination with a control having a relatively large amount of static friction, the result is at best unpleasant and at worst little short of dangerous.

Upwards of 30,000 miles driving with cars having these unfortunate dual characteristics leaves a thoroughly bad impression, overshadowing many virtues of otherwise satisfactory vehicles. The large dissatisfaction caused by such a detail may appear out of all proportion, but none the less it will remain one of the lasting impressions of the cars in question. The jerky progress in thick traffic when "inching" along or in short bursts of acceleration in low gear is singularly disturbing for the passengers, while the driver must unnecessarily slip the clutch to avoid ramming the car in front. His mental anxiety is not eased by the knowledge that reasonable pressure on the pedal will probably have no effect, and the engine therefore stalls as the clutch is engaged. Pressure adequate to overcome the friction in the throttle control will result in handsomely overshooting the desired throttle opening. In either case smart work with the clutch is essential to avoid disaster. Experience has shown that these conditions are the best that can be expected, even when the mechanism is in good order.

Lubrication

Since no provision is made for lubrication there is no alternative, after a few thousand miles, but to strip the cable from its housing and remove the accumulated paste of partly dried oil and fine dust that is gradually promoting final seizure of the cable. Ultimately a climax is reached at about 15,000 miles when "whiskers" appear on the cable, owing to wear of the individual strands, thus rendering operation of the throttle quite intolerable, with the added possibility of danger from a throttle that may stick open. Replacement of the entire assembly then becomes essential.

In some designs the cable runs in a curve that is either continuously upward or continuously downward. When this is so, it is possible to lubricate from one end without complete detachment. When, however, the cable runs both up and down, the actuation soon needs complete dismantling. Free oil still refuses to run uphill and there is no alternative to stripping down. In such cases the provision of even a hole at the highest point, would be of some assistance.

It may be argued that this method of throttle operation is cheap and simple. Cars of comparable price, however, embody well thought-out linkage, using rods and joints suitably located to accommodate movement of the engine on its mountings and having far less static friction. It

is not an insuperable problem, therefore, to provide a sound, engineering linkage, and the possible small increase in cost is negligible when offset against driver irritation, and consequent loss of reputation and goodwill. Doubtless the flexible cable has many virtues in suitable circumstances. The automobile throttle, however, is not to be numbered among them, and indeed it would be extremely difficult to find a less appropriate application.

The question of pedal ratio, although rather more subtle, has been solved many times, and failure to arrange for a slow opening of the throttle off the idle setting is merely a reffection on the quality of the initial layout work in the drawing office. Even a mechanical system that appears correctly proportioned on the arrangement drawing may fall short in practice because an essential member in the linkage has been overlooked. The lever system begins at the driver's heel, and neglect of the foot as part of the system may negative the best intentions of the designer. Seating position and height, and the angle of the toe-board, are all factors to be taken into consideration. Designers of these detail points often fail by concentration on the immediate surroundings to the exclusion of the whole.

One matter affecting driver comfort in which body designers appear to be travelling definitely backwards rather than forwards, is in the matter of seat design. The modern broad, deeply padded seat, while comfortable regarded purely as a static chair, is of no use as a position from which to drive and control the motor car. Basically, the fault is that the modern seat leaves the driver much too detached from his vehicle. A good driving seat should be reasonably well padded, but above all, should quite firmly contain and locate the driver. This applies not only to the back and shoulders, but to the hips and upper part of the legs as well. The "bench" seat, even with armrests, is an abomination.

The bucket seat

Many of the older cars had properly designed "bucket" driving seats. Even the drawbacks of the vehicles themselves in the matters of steering and suspension, were much mitigated by the fact that the driver was able to exploit to the full whatever favourable characteristics the car possessed. The advent of much faster vehicles in the shape of family saloons with sports car performance, brings this matter very sharply into the forefront of essential requirements. It is high time that body designers began taking an entirely different view, at any rate of driver accommodation. We have now reached a stage when it is absolutely essential that much more vehicle control should be built into the driver's seat itself.

This driver viewpoint is also in some degree condemnatory of the steering column change gear lever. While convenient in many ways, this device can, by its slowness, be a continual source of irritation to a keen driver. However well it is assembled and adjusted, it cannot embody the rigidity, and consequent rapidity of action, with the positive "feel" of the gear lever working directly on the striking fork. It is to be hoped that cars will still be made in which the old method of actuation is adhered to, particularly when they are of sports, or near sports character and are intended to provide driver satisfaction as apart from mere transport.

Dust sealing

Dust sealing is a matter that has recently been receiving a considerable amount of attention. Until wider experience has been obtained with the modified arrangements embodied in more recent models, comment would be premature. Something may be said, however, concerning the method of sealing hinged quarter windows and ventilating flaps. Contrary to what it has become reasonable to expect, in some cases the sealing at these points is so effective that the window cannot be opened without dragging the rubber seal from its housing. Exposure to sun effectively bonds the rubber and glass even when the heat is not excessive.

If the car has been parked in the sun for any length of time, any attempt to open the ventilators necessitates a struggle during which the operating mechanism is overstrained. Actually, in such cases a sharp blow on the glass is usually necessary before any impression can be made, followed by a period of tucking the seal back into its channel.

If this can occur in normal British summer weather the effects of warmer climates can well be imagined.

Thoughts on the subject of ventilators must inevitably be related to the question of heating and ventilating. This, however, is too large a subject to embark upon at this juncture. Attention may, however, be drawn to the outstanding error of taking heater air in from a duct placed low down at the front of the car. In thick traffic this can be a quite serious matter if the driver is not alive to what is taking place. Even in moderate traffic, a sufficient volume of fumes will be drawn into the car to give the impression that the engine fumes of the car itself are making their way in. Neither does there appear to be any simple solution to the problem of intake location. The sides and even the rear of the car are open to objection on the same grounds. Any intake, in fact, below the waistline will be productive of bad air within the body.

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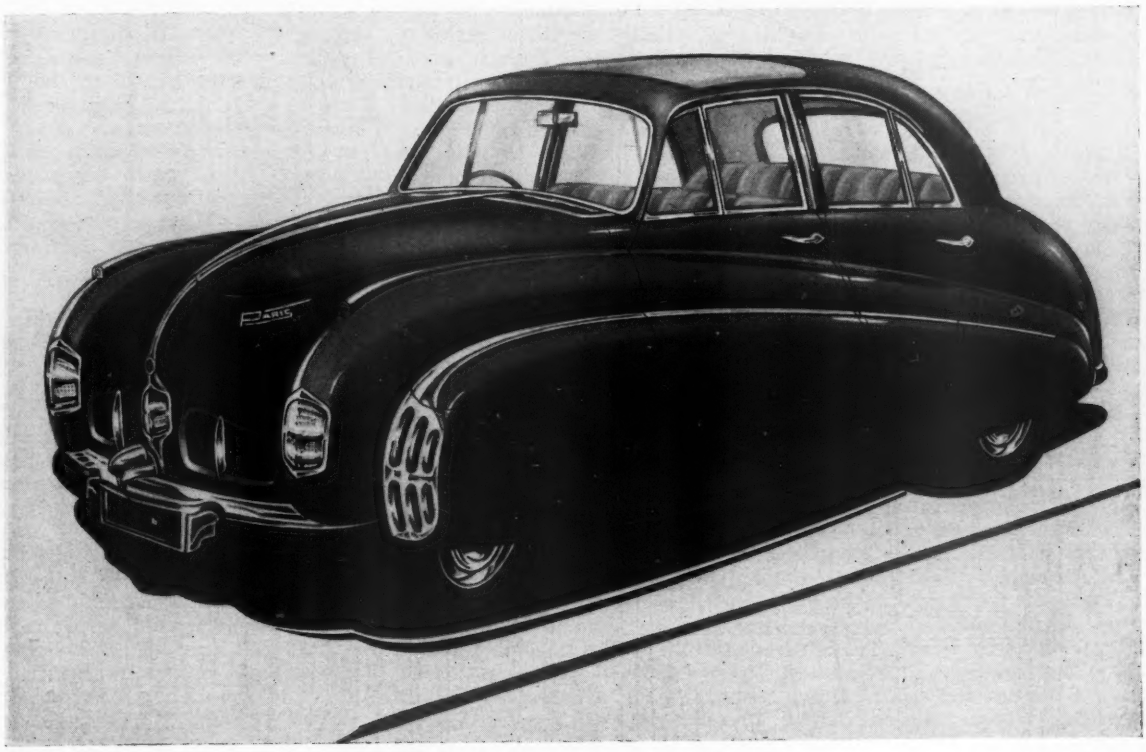
BODYWORK COMPETITIONS

A Review of the Exhibits Submitted in the I.B.C.A.M. Contest



Four-door four-light saloon by P. J. Wharton.

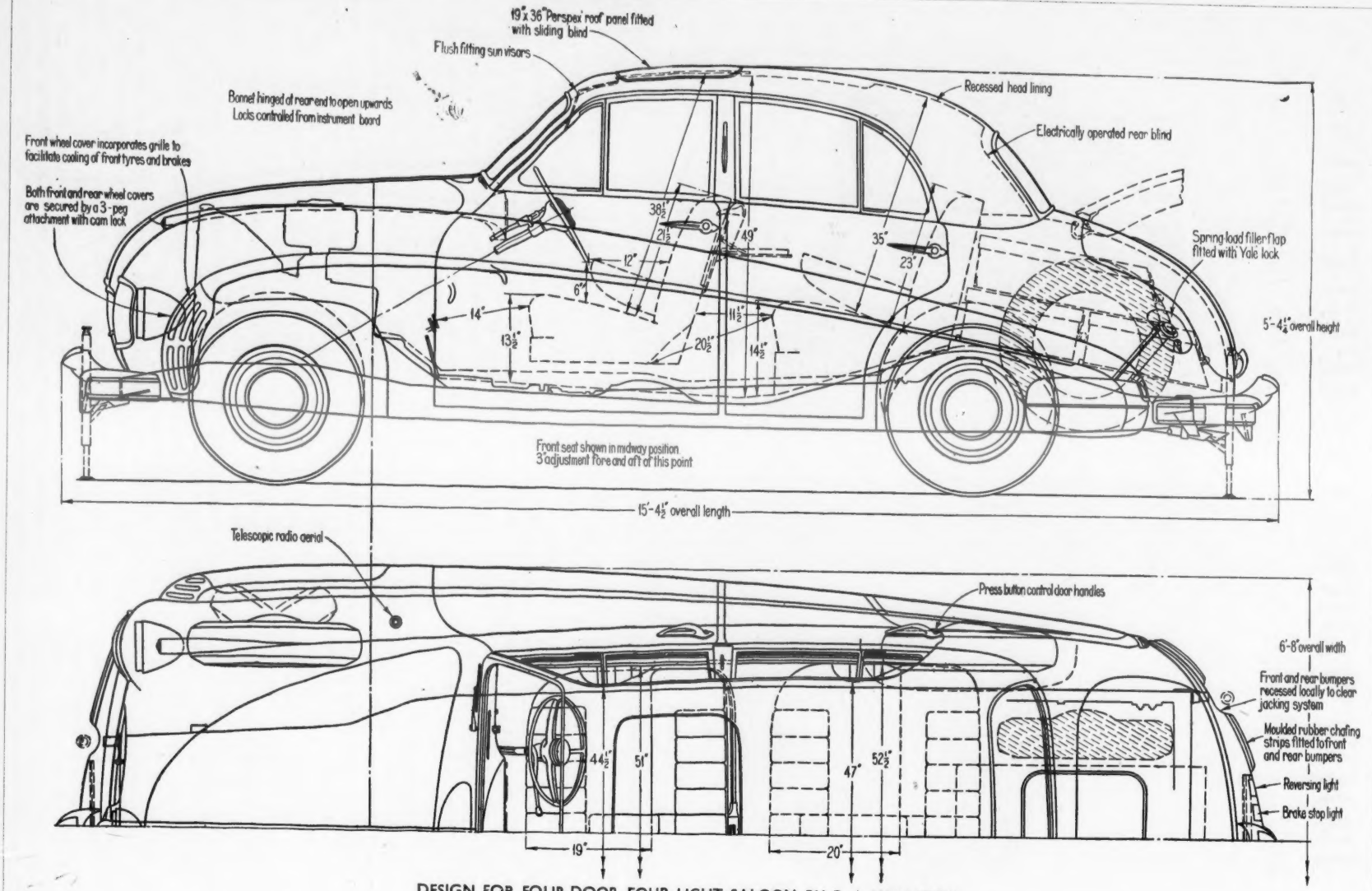
AN annual event that maintains its interest for coachwork draughtsmen and others concerned with body design and construction, is that organized by The Institute of British Carriage and Automobile Manufacturers in association with the S.M.M. and T., The National Federation of Vehicle Trades, and The Worshipful Company of Coachmakers and Coach Harness Makers of London, for drawings of all descriptions. This competition covers various classes of road vehicles, ranging from medium-sized saloons to public service vehicles. Most sections were very well supported, and some fine entries were to be seen. An outstanding feature was the general tendency in the private car classes for competitors to follow the prevailing trend for full width bodies in which the front wings are carried back to form an integral part of the lower body panels. Although support for the principal classes was good, it is to be regretted that Competition 2 attracted only one entry, and on this account the judges made no award. In this particular section, competitors were asked to submit their ideas on two of three subjects, the first of which was devoted to air conditioning, and details were required to show the method of air entry, circulation, extraction, draught exclusion, and temperature control. The troublesome questions of demisting and defrosting were also to be covered. The second portion of this class asked for a seating layout of a six-seater saloon car, whilst the third alternative dealt with door hardware, competitors being required to illustrate door hinging, window regulation, door locking and the means recommended for checking the doors in the open position.

It is to be hoped that the organizers will not be deterred by the lack of interest shown, for it is felt that this is a most valuable section, particularly those portions of it which deal with ventilation and door hardware, both being items which are of current interest.

Competition No. 1

This was open to persons of British nationality, without age limit, and was for an outline drawing in ink of a four-door, four-light saloon. Entrants were supplied with a coachbuilders chassis drawing and were required to produce an outline drawing to a scale of 1½in to the foot, showing seating dimensions, spare wheel and luggage accommodation. A perspective coloured drawing was also required, and competitors were at liberty to use hand-brushed or mechanical means for the finishing of this drawing. Full scope for styling and originality of design was provided by the fact that the body could be treated as one for an individual "one off" order.

The first prize of £50 and the company's medal was awarded to P. J. Wharton for a well executed drawing of a modern style of saloon body. Both the outline drawing and the smaller coloured drawing show excellent work, and this competitor has had the forethought to include two


19" x 36" Perspex roof panel fitted with sliding blind
Flush fitting sun visors
Bonnet hinged at rear end to open upwards. Locks controlled from instrument board
Front wheel cover incorporates grille to facilitate cooling of front tyres and brakes
Both front and rear wheel covers are secured by a 3-peg attachment with cam lock
Recessed head lining
Electrically operated rear blind
Spring load filler flap fitted with Yale lock
5'-4¼" overall height
38½"
21½"
49"
35"
23"
12"
6"
14"
13½"
11½"
20½"
14½"
Front seat shown in midway position 3" adjustment fore and aft of this point
15'-4½" overall length
Telescopic radio aerial
Press button control door handles
6'-8" overall width
Front and rear bumpers recessed locally to clear jacking system
Moulded rubber chafing strips fitted to front and rear bumpers
Reversing light
Brake stop light
44½"
51"
47"
52½"
19"
20"

DESIGN FOR FOUR-DOOR FOUR-LIGHT SALOON BY P. J. WHARTON.

small outline drawings offering alternative colour treatment. His design lends itself to a dual colour scheme, and one of the supplementary sketches shows a treatment in which the lower portion of the wing and body side panel, together with the roof, are carried out in a darker shade than the remaining body panels. A second suggestion reverses this treatment, whilst the coloured drawing taking a three-quarter front view, is in blue. It is to be noted that this competitor is one of several who favour this particular viewpoint, and the coloured drawing includes a diagram showing the geometric layout of the perspective.

General amenities

Concerning the design itself, a bench type front seat is used, and both front and rear seats are equipped with folding armrests. Further comfort is provided by fixed armrests on the doors, and it is noted that the centre armrest in the rear seat has a compartment with a sliding lid for the reception of small articles. It is noted that in this design, considerable thought has been devoted to the provision of pockets and similar receptacles for the stowage of small personal items. For example, in the centre of the front seat back there is a cabinet fitted with a roller shutter, embodying also an ash tray for the use of rear seat occupants. Folding tables also are arranged in the back of the front seat, whilst the near side inner scuttle panel accommodates a folding tool tray containing the usual small tools. The specification of this body includes a telescopic aerial housed in the off-side front wing, press-button locks, electrically operated door glasses, and a perspex roof panel over the front seats. The luggage accommodation is somewhat restricted by the vertically mounted spare wheel, but nevertheless the lift-up boot lid gives access to four fitted suitcases that should accommodate reasonable luggage for the passengers in a car of this size.

Both doors are hinged on the forward pillars and the windows contain the usual ventilating panels. There is a curved screen; and the bonnet panel, which is hinged at its rearward end and is locked from within the car, has a well rounded frontal aspect that harmonizes agreeably with the front wing treatment. Detachable wing spats are a feature of this design and those concealing the front wheels extend on to the forward face of the wing and combine a grille panel permitting the free entry of cool air to the front tyres and brake drums, a commendable feature. This competitor is to be congratulated on his excellent draughtsmanship and obviously has a fine appreciation of the value of well-drawn interior views.

A second prize of £25 and the company's medal was awarded to L. J. Stevenson, whose attractive design is reminiscent of prevailing American tendencies. This impression no doubt is created by the three-piece rear light which extends on to the quarter panels, curved glass of course being used for this rear light, and also for the fixed windscreen. Another feature which associates this design with American practice is the treatment adopted for the radiator grille, which in its centre houses a pass lamp, whilst the headlamps are built into the forward extremities of the wings and embody the side lamps.

Seating accommodation provides a bench type front seat which has a folding centre armrest that for the driver's convenience is offset to the near side, this obviously being necessary when the position of the steering wheel in relation to the seat is considered. There is also a wide folding armrest to the rear seat, whilst the door armrests are of the type that can be used for door pulls. Both interior and exterior door handles are of the press-button pattern and the outline drawing clearly indicates the positioning of door hardware. The luggage compartment is of exceptional length and has a floor dimension of 50½in from the base of the seat squab board to the extreme end of the floor, and it could be imagined that with this extremely long tail loaded to capacity the effect on the handling of the car might be unpleasant. The lift-up boot lid is carried on double links, and the shut line is arranged to follow a curved deck line.

Regarding the length of the luggage compartment, it is noted that although the wheelbase of this chassis is 105½in the overall length of the car is 190in, the overhang behind the rear wheels being half as much again as that forward of the front wheels. This competitor's drawings include a coloured drawing in excellent style, in this case also blue being the selected colour treatment.

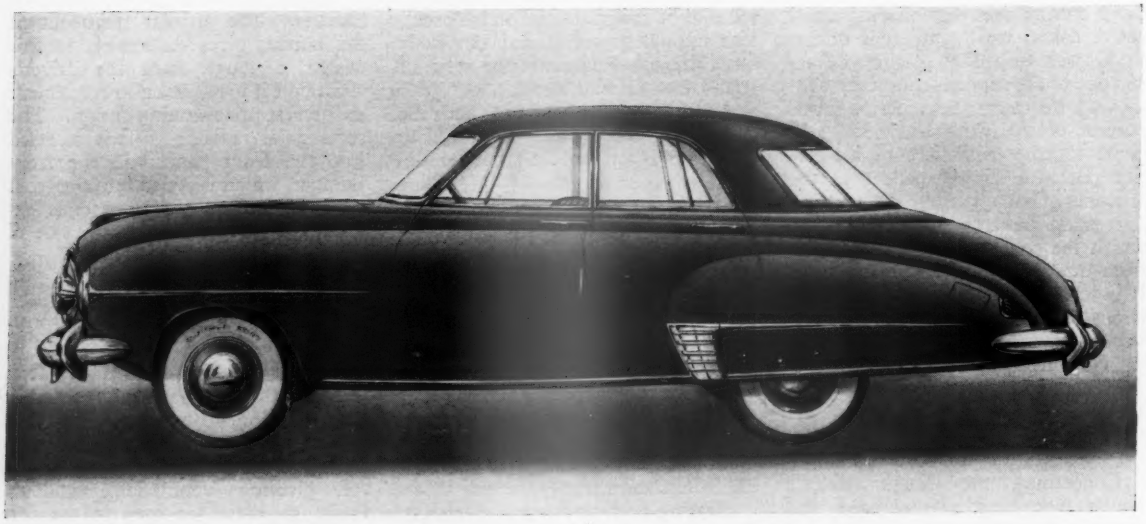
A third prize of £15 and the company's medal was awarded to C. G. Neale, whose coloured drawing is outstandingly good, but shows less detail than other prizewinning competitors in his outline drawing. A feature of this design is the somewhat angular treatment of the bonnet and luggage boot panels, and this styling is supported by a raised section formed in the front wings and door panels to accommodate angular shaped side and rear lamps, and also housing the door handles. The angular treatment of the bonnet gives distinction to this design, although from the driver's viewpoint it may not be so convenient as the first prizewinning design. The rectangular radiator grille is recessed into the front panel and the front bumper, which is styled into the front panel formation, has deep ends extending well round on to the side of the car.

Ventilation

Concerning the body itself, the screen has ample curvature and is divided by a central channel. It is obvious that the angle of the screen and consequently the shape of the forward ends of the front door windows, precludes the convenient use of pivoting ventilation windows, although these are used in the rear doors. The front doors, therefore, are equipped with externally mounted glass panels or wings that can only act as deflectors, and therefore will only extract air from the interior. It is generally conceded that ventilating windows should now be made to open beyond the 90 deg angle and can then be made to feed fresh air into the front compartment, this feature being largely influenced by the desire to satisfy overseas customers.

Access to the luggage compartment is through a horizontally folding lid, and it appears that this lid is to be of sufficient thickness to accommodate a spare wheel, an internal lid giving access to this wheel and also to the tools. It is felt, however, that such an arrangement would not meet with general approval, for substantial hinging arrangements are needed and a counterbalancing device is regarded as essential. The outline drawing gives no information on these points and the drawing in general lacks the necessary detail work that other competitors have shown.

For his coloured drawing, H. G. Poole was highly commended and an award of £5 was made. Certainly this coloured drawing is outstanding and the detail work is exceptionally good. The only point on which this drawing can be criticised is in the treatment of the interior seats, although obviously the effect of reflected light on the door and screen glasses would to some degree give an indefinite outline to the trimming. This competitor's design follows the customary full width style, with bonnet and luggage boot panels having very similar formation. The head and side lamps are housed in the forward extremities of the front wings and the chromium plated radiator grille is recessed into the upper portion of the bonnet front cowling.



American influence is prominent in L. J. Stevenson's four-door saloon.

Beneath this grille there are three louvred inlets for air, whilst the upper panel of the bonnet is provided with what appears to be a form of ventilator giving an extractor effect, this being situated immediately above the top header tank.

An award of £5 was made to P. M. Finch, who was highly commended for his outline drawing, which featured a saloon following the razor-edge style. This is not a particularly attractive design, but this competitor has obviously given considerable thought to his subject. The construction of the body features light alloy members formed in 0·048in Birmal sheet, panelled in mild steel. The front wings are in light alloy and are made detachable, whilst the rear wings are integral with the body and have a wheel spat held in place by three sprung arms.

This competitor clearly indicates the arrangement under the rear end of the bonnet, which houses the heating equipment, windscreen wiper and bonnet hinges, whilst the air circuits created by the heating and demisting equipment are clearly shown. It is obvious that this outline drawing compares very favourably with those of the other competitors, and it is unfortunate that his coloured drawing lacks the finish which would undoubtedly have gained a higher award.

Amongst other outstanding drawings were those shown by "Envoy" and "Mercury," both of whom did not figure in the prize list. "Envoy's" coloured drawing shows an attractive but conventional design, with the front wings carried through high on the body side and having a lower slightly offset panel that is partly carried on to the front panels. The upper portions of the wings appear to grow out of this base, which in the coloured drawing are shown in a darker shade of blue. The outline drawing merely shows the seating arrangements and general interior dimensions and is completely lacking in other detail work.

The coloured drawing entered by "Mercury" is an attractive piece of work, although the design of the body, due to the massive treatment of the bonnet, is not pleasing. The distance between the front wings is spanned by a panel which is curved in plan and embodies a series of horizontal chromium plated mouldings. Unlike most other competitors, this entrant allows his front wings to fall as a pass towards the rear, and the rear wing, although forming a portion of the rear door panel, appears as a separate unit. In this case also the competitor gives little more than the seating arrangement and dimensions on his outline drawing.

Competition No. 3

This was open to students of any technical school or employees under the age of 21, and it was also possible for Service men to enter if they were under the age of 23 at the closing date for the receipt of entries, these Service men having served not less than twelve months in the Forces. Competitors were required to submit an outline drawing of an open utility body, having accommodation for one or two passengers in an enclosed driver's compartment in addition to the driver. The rear portion of the body was to be open and to be provided with optional weather protection, the loading capacity of this vehicle being 5cwt. Competitors were provided with a coachbuilders chassis drawing, and the outline drawing was required to a scale of 1½in to the foot, and was to be in pencil showing the side elevation, half front, half rear, and half plan views.

The first prize of £15 was awarded to D. B. Jennings, whose design follows attractive lines and embodies many of those ideas which are normally associated with private cars. For example, the front wings are extended rearwards to form outstanding portions on the front doors, and an opportunity is taken of hinging the doors on the vertical line which this treatment provides. The door handles are of press-button type and the driver's cab is equipped with pivoting ventilation windows.

In general outline, this design closely resembles that of a coupé, for the cab roof is swept downwards immediately behind the driver's seat to merge into a swept side panel which follows a gradually falling line towards the rear end of the chassis. A feature of the goods-carrying portion of this body is the steel roller shutter, which gives optional weather protection, and this shutter is contained on a roller housed in the cab quarter panel behind the driver's seat. A hinged door behind the driver's head gives access to the shutter mechanism, and the only fault that can be found with this arrangement is the possibility of water finding its way between the slats of the steel roller shutter and causing corrosion.

From this drawing it is obvious that to obtain the low floor line in the goods-carrying portion, some slight re-arrangement of the spare wheel is necessary, for the floor is shown only by one line that coincides

with the upper face of the spare tyre. There seems no reason, however, why this spare wheel should not be farther back, and since it is slightly inclined, sufficient room would no doubt be available. Another detail that calls for criticism is the lack of a rear light to the driver's cab, this being due to the roller shutter arrangement which completely blocks rearward vision. The competitor should, therefore, have indicated an external mirror. Running boards are concealed and the rear wings are fitted with detachable valances, and although no constructional details are shown, it is stated that the pressed steel method is to be followed.

The second prize of £10 10s was awarded to D. H. Smith with an attractive design embodying some novel features. Weather protection for the goods-carrying portion is provided by a tilt supported by a linked head fitting, and although full details are not shown, a note indicates that the truck has four possibilities, for in addition to its use with the head open and closed, it can also be used with a detachable gantry composed of tubular alloy, and also with detachable hay racks, also to be constructed of light alloy.

In addition to the cab seat, which will accommodate three persons, there are also removable bench seats for five passengers in the rear portion of the body. Composite construction is specified and the equipment of the body includes quick-lift windows and hinged ventilating panels in the front doors. This competitor, too, favours the type of front wing that is carried over the front doors and forms an integral part of the side panels, there being no visible rear wings.

The third prize was awarded jointly to C. A. Lodge and M. J. Knight, both of whom received £3 15s. The drawing submitted by C. A. Lodge indicates a conventional type of utility body, in which the goods-carrying portion has weather protection formed by a canvas covering on a tubular framework. This competitor also carries the front wings on the body side panels and a well curved bonnet and screen are a feature of the design.

The entry submitted by M. J. Knight differed from other prizewinning designs in this section, in that the method used to give protection to the goods-carrying compartment does not rise above waist height, and consists merely of a tarpaulin covering secured by eyes at either side. This, of course, somewhat reduces the covered capacity of the body, and may have influenced the judges in their decision. The body style itself is attractive, and in this case the front wings which pass across the front doors, follow a falling line which terminates just behind the rear wheel centre. A perspective sketch showing the interior of the cab is included.

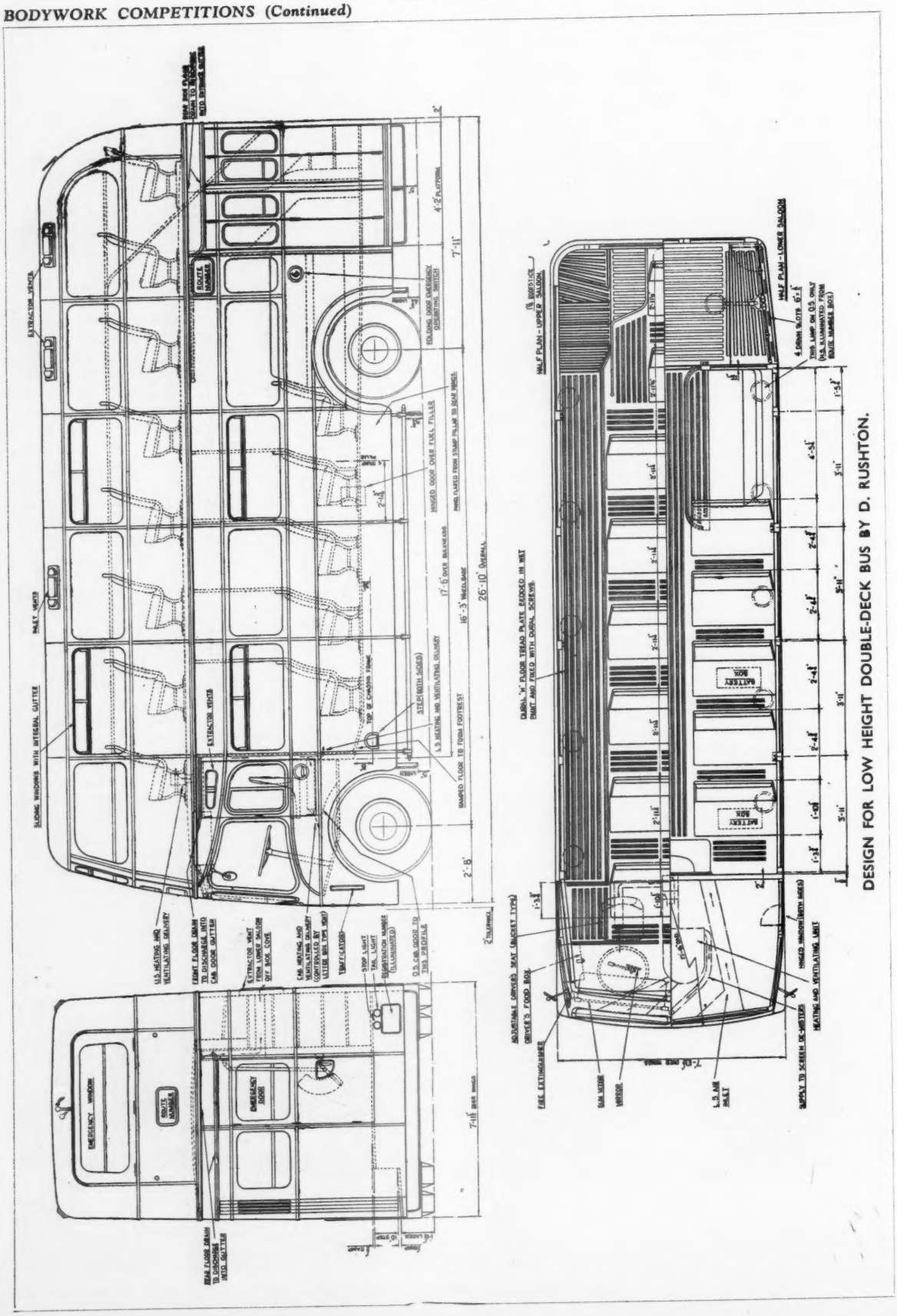
Competition No. 4

This competition called for a general arrangement drawing in ink, of a low height double-deck bus suitable for operation in this country. The height of the body above the frame was not to exceed 11ft 6in, and other dimensions specified were an overall length of 27ft and a width of 8ft. Competitors were asked to give special consideration to the driver's comfort and visibility, and the ventilation of the cab was also a special requirement. The drawing was required to a scale of $\frac{3}{4}$in to the foot and was to be in ink, the usual views being shown, whilst details of construction to a scale not exceeding 1$\frac{1}{2}$in to the foot were required. Competitors were supplied with a coachbuilders chassis drawing.

The first prize of £75 and the company's medal was awarded to D. Rushton, who has obviously given considerable thought to the preparation of his design. The seating arrangements comprise a central gangway style in the lower saloon, giving accommodation for 26 passengers, whilst in the upper saloon the gangway is arranged on the off-side. The upper saloon gives accommodation for 30 passengers, and all the seats in this compartment except the last two, are of the four-passenger type. The lower saloon has a clear height of 5ft 11$\frac{1}{8}$in, whilst in the upper saloon there is a clear head room of 5ft 8$\frac{1}{2}$in, measured at the gangway. The upper saloon seating floor is 11$\frac{1}{4}$in above the side gangway floor, and a point worthy of note in the seating layout of this body is that the upper saloon four-passenger seats are each equipped with two lower saloon two-passenger cushions and one pair of

Angular treatment in C. G. Neale's four-door saloon.

BODYWORK COMPETITIONS (*Continued*)


EMERGENCY WINDOW
EMERGENCY DOOR
ROUTE NUMBER
EXTRACTOR VENT
SLIDING WINDOWS WITH INTEGRAL GUTTER
H.S. HEATING AND VENTILATING DELIVERY
LIGHT FLOOR DRAIN TO DISCHARGE INTO CAB DOOR GUTTER
EXTRACTOR VENT FROM LOWER SALOON INTO SIDE COVE
H.S. HEATING AND VENTILATING DELIVERY (CONTROLLED BY LOUVRE BOX TYPE VENT)
TRAFFICATORS
STOP LIGHT
TAIL LIGHT
REGISTRATION NUMBER (ILLUMINATED)
O.S. CAB DOOR TO TIRE PROFILE
REAR FLOOR DRAIN TO DISCHARGE INTO GUTTER
7'-11" plat mean
EXTRACTOR VENT
H.S. HEATING AND VENTILATING DELIVERY
STEPS (WITH DOORS)
TOP OF CHASSIS FRAME
RAMPED FLOOR TO FORM FOOTREST
2'-8"
17'-6" OVER BRACKETS
16'-3" WHEELBASE
26'-10" OVERALL
HINGED DOOR OVER FUEL FILLER
PANELLING FROM STAMP PILLAR TO DOOR HINGES
FOLDING DOOR EMERGENCY OPERATING MECHANISM
4'-2" PLATFORM
7'-11"
ROUTE NUMBER
EMERGENCY DOOR
EMERGENCY WINDOW
HALF PLAN - UPPER SALOON
HALF PLAN - LOWER SALOON
4 DRAIN SLOTS 6" c.f.
TAIL LAMP ON O.S. ONLY (N.S. ILLUMINATED FROM ROUTE NUMBER BOX)
DURAL 'M' FLOOR TREAD PLATE RECESSED IN MAT PAINT AND FIXED WITH DURAL SCREWS
BATTERY BOX
BATTERY BOX
ADDITIONAL ARMCHAIR SEAT (ANCHET TYPE)
DRIVER'S FOOD BOX
FIRE EXTINGUISHER
SUN VISOR
MIRROR
H.S. AIR INLET
HINGED WINDOW (WITH ARMS)
HEATING AND VENTILATING UNIT
SUPPLY TO SCREEN DEMISTERS

DESIGN FOR LOW HEIGHT DOUBLE-DECK BUS BY D. RUSHTON.

two-passenger squabs, this arrangement, of course, greatly simplifying servicing and replacement.

The body is of all-metal construction and it is intended that the rear end should be self-supporting, requiring no frame extension for its support. All rivet and bolt holes are to be jig drilled, and as far as possible components have been made interchangeable. This competitor submitted a specification which makes a special point of metal treatment to avoid corrosion. For example, all contact surfaces are to be painted immediately before riveting or bolting, Carbo-lastic compound being used for steel joints, whilst di-electric paint is specified for steel and aluminium joints. Steel self-tapping screws are to be tinned and all wood-screws and bolts less than $\frac{3}{8}$in diameter are to be electro-zinc plated. It is also mentioned that the timber used is to be of first quality and its moisture content is not to exceed 15 per cent.

Heating and ventilating

In the matter of heating and ventilating, a unit is fitted in the cab and is attached to the roof. The unit is covered by a detachable asbestos-lined silencing panel and the air enters the unit by a duct extending from a louvre above the off-side screen and having passed through the heating unit, is discharged from a further duct at the rear of the unit. The supply duct is divided, one portion being led through the floor into the upper saloon, and is discharged from beneath the front seat, whilst the other branch feeds heated air into the lower saloon, being released in the centre of the bulkhead structure. A branch is taken from the lower saloon duct into the cab.

The rear entrance platform is equipped with double folding doors and these are air operated, being under the control of the driver. The door engine is housed in a box below the platform roof boards.

The second prize of £37 10s and the company's medal was awarded to R. Parsons, whose design featured a 51-passenger metal framed body, mainly composed of high duty aluminium alloy with timber inserts to carry the panels. Aluminium is also used for the panels and vertical joints are covered by light aluminium moulds. In this design the entrance door is situated forward of the wheel arch and is fitted with a power-operated sliding mechanism, which is under the control of both driver and conductor. The staircase is at the front immediately behind the bulkhead and runs from the near-side to the off-side, giving convenient access to the upper saloon gangway which is on the off-side.

Fresh air is provided in both saloons and cab by means of a heating and ventilating unit, which is situated under the cab roof, fresh air for this heater being admitted through a grille above the centre of the screen. As desired, heated or cooled air is delivered to the saloons by ducts at floor level, whilst air is extracted by means of ventilators on the upper saloon roof. In the lower saloon there are extractor vents above the windows.

This competitor supplied a diagrammatic layout of the electro-pneumatic door control and push buttons, whilst the layout of the heater system is clearly shown. There is also a pictorial view of the sliding door gear, and in addition to the very well defined layout drawings, a perspective view gives a good impression of the finished vehicle.

The third prize of £15 and the company's medal was awarded to F. J. C. Hollyman with a 57-seater design, constructed in aluminium alloy. The lower saloon is constructed in seven main units, consisting of the under-frame, front bulkhead, rear bulkhead, rear frame, side frames, cab and intermediate roof. For the construction of the upper saloon there are four main units, the side frames, two ends, and main roof. The saloons are panelled and completely finished independently and are then assembled by bolting through the lower saloon cant rail and upper saloon sill angle, both of which are continuous all round the body. A cover strip is then pop-riveted on to close the joint on the exterior of the body.

This competitor utilizes a rear platform entrance, which has single-fold doors that are hand-operated. Concerning the seating arrangements, the upper deck is planned to give a row of seven single seats against the off-side panel, whilst a row of eight seats, each seating three, against the near-side panel gives a total upper saloon capacity of 31 seats, this layout, which has a slightly off-set gangway, being claimed to give a more balanced loading than the orthodox type.

A drawing that was highly commended and for which the entrant, S. H. G. W. Anderson, received an award of £10, gave a seating capacity of 60 persons, 27 in the lower saloon and 33 in the upper saloon. The entrance door, which is of the folding type, is on the near-side immediately behind the bulkhead, and gives access to a transversely arranged staircase. In the upper saloon the gangway is on the near side. This competitor does not give such detailed constructional information as the more successful entrants in this class, but his half cross-section shows careful thought and is identified by numbers with a tabulation giving the specification of the materials used.

Competition No. 5

In this competition a general arrangement drawing of an ambulance was required, and this vehicle had to be suitable for a general hospital dealing with sickness and casualty cases. Accommodation was to be provided for two stretchers and for four sitting cases when the stretchers are not in use, additional space being available for the attendants. The drawing was required to a scale of 1½in to the foot and was to be in ink, showing the usual views.

The first prize of £50 was awarded to B. J. Garner, whose interior layout embraced an attendants seat in the front cab, with a nurse's seat on the near side immediately behind the front bulkhead. An interesting feature of this body is the isolation method of body mounting that is used, and this competitor gives a detailed view of the Metalastik conical mounting which is bolted to short outrigger brackets extending from the chassis frame. The drawing shows the manner in which the supporting arrangements of the stretchers can be utilized to accommodate the sitting patients, the seat cushion normally remaining beneath the stretcher, which when removed, enables the cushion to be brought forward and to rest upon the inner runner. The outer runner accommodates the base of the squab which since the top of the squab is hinged and can on occasions be pushed back against the inner panel, provides sufficient room for the stretcher.

The second prize of £25 was awarded to S. Reynolds, whose design is characterized by flowing mouldings on the lower side panels. In this design the single piece front seat accommodates the driver and one attendant, whilst a seat for a nurse is provided immediately behind the bulkhead on the off-side panel.

The third prize of £15 was awarded to W. Back, whose stretcher supporting arrangement utilized the folding squab method, operated by a quadrant mounted on the end of the stretcher support bolts. The inner track taking the stretcher is mounted on a slide and when not required, is slid outwards into a recess formed in the front edge of the longitudinal cushion. This competitor clearly illustrates his method of body construction, which uses timber framework, and full details are given of the folding rear step arrangement.

STEEL CASTING PRODUCTION

A Survey of the Methods Employed by Leyland Motors, Ltd.

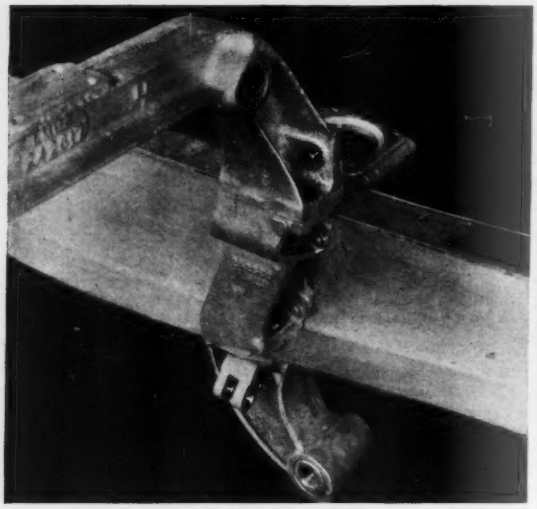


Fig. 1. Steel castings for the engine rear cross-member mounting on a Leyland chassis.

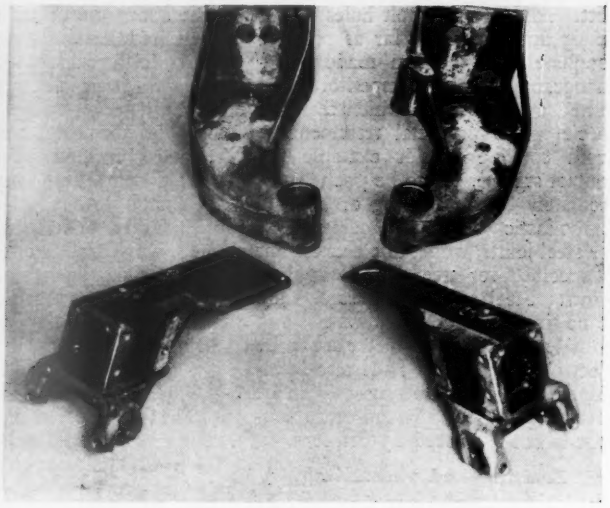


Fig. 2. Right- and left-hand front spring brackets made from steel castings.

PRESENT-DAY steel castings are much better products in every way than those of pre-war days. In the main this can be attributed to two factors. First, extensive research has been carried out on the problems of steel casting, and the knowledge so gained is now being employed. Secondly, the equipment used in the production of the castings is greatly improved. In particular, important advances in heat treatment technique guarantee that every product is fit to satisfy the requirements for its particular function, especially with regard to stressing. This is of importance in the construction of commercial vehicles, which carry heavy loads and are often subject to violent shock stresses when travelling on poor road surfaces.

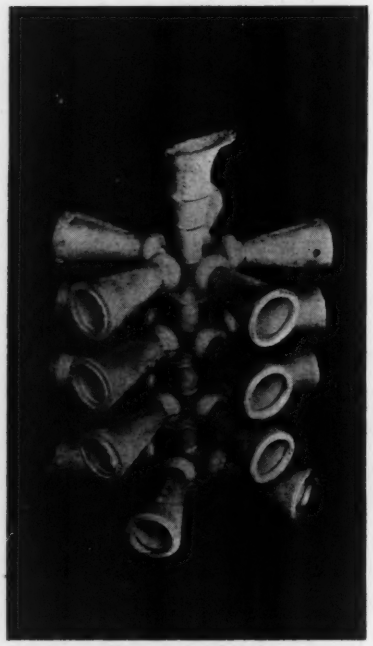


Fig. 3. Body mounting brackets centrifugally cast 20 at a time.

For commercial vehicle work, lightness and low cost are essential, and here steel castings can play an important part. Under modern foundry conditions it is a relatively simple matter to produce steel castings which are so accurate that the machining allowances can be reduced considerably below those previously considered necessary. Such castings use the minimum amount of metal. Furthermore, the high strength/weight ratio of cast steel makes possible the use of thinner sections than can be employed with any other metal or process. This, in conjunction with improved accuracy, ensures that the components are of minimum weight. In turn this makes for cheapness, since it increases the number of castings that can be produced from a given weight of metal. It is also important that the smaller machining allowances reduce the time taken for machining which is often a considerable item in total costs.

In order to obtain greater weight reduction and to reduce costs and times further, certain parts may be fabricated by welding. For inclusion in such work the steel casting possesses an advantage because of its particular suitability for this type of construction. It allows complicated or heavily stressed parts to be made as castings and afterwards welded to other more lightly stressed units.

Chassis units

Naturally, steel castings play an important part in the design of heavily

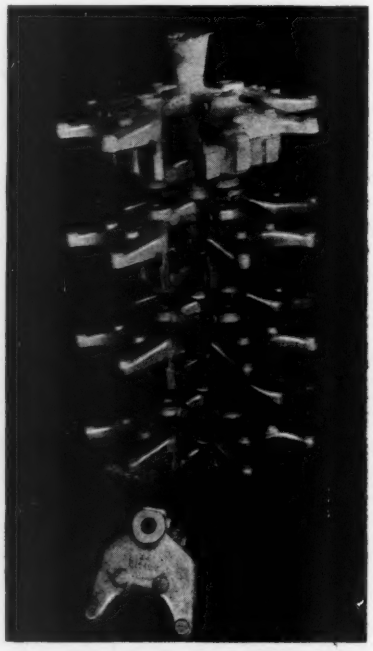


Fig. 4. Spring brackets, each weighing 18 lb spun cast 16 at a time.

Fig. 5. A single spring bracket as originally produced in normal sand moulds.

Fig. 6. Brackets, similar to that in Fig. 5, produced in pairs by centrifugal casting.

stressed engine components, but their use in chassis construction is of particular interest, as will be seen from the following details referring to heavy-duty vehicles built by Leyland Motors, Ltd. This firm manufactures a wide range of commercial chassis for the carriage of both passengers and goods, and approximately 118 different steel castings are required for the current range of vehicles. Naturally, the number per complete chassis varies considerably from model to model. They are employed at all positions subjected to heavy loading or liable to sudden stressing.

A typical example is provided by the engine rear cross-member mounting (Fig. 1). Here, the weight of the engine is supported by two cast steel members, one of which is seen in the illustration; this rests on the chassis side frames and also on a cast steel support bracket, to which it is secured with bolts. The support bracket fits snugly inside the side frame, where it is bolted in position. Below are the cast steel engine torque stay bracket and the front spring rear bracket, and on the outside of the frame is a body mounting bracket, also a steel casting. The five castings form an important assembly which is subjected to most of the loading and stressing encountered by the vehicle in service. Rear and front engine mounting for another chassis are shown in Fig. 2.

Foundry methods

Leylands Motors, Ltd., have produced steel castings since 1918, and the present equipment is of very modern design, including two electric arc furnaces operating on the basic process. Much of the work is cast in orthodox types of sand moulds, although, as will be mentioned later, the centrifugal process is also employed. The general run of castings range from 1½-120 lb weight.

A considerable amount of steel swarf and scrap is available from the company's large machine shops, and this material forms the basis of the furnace charges from which steel for castings is made. As a result of the steel-making practice and heat treatment employed, very sound castings are obtained which have an ultimate tensile strength of 30-40 tons/sq in and exceptional toughness.

Heat treatment comprises oil quenching from 900 deg C. and tempering from 600-650 deg C., this giving steel with the following average mechanical properties:—

Maximum stress, 35-40 tons/sq in.
Yield point, 25 tons/sq in minimum.
Elongation, 25-30 per cent.
Izod impact value, 40-90 ft/lb.

The equipment includes gas-fired furnaces with automatic temperature control; these, together with the quenching tanks, etc., are arranged in-line so that loading, unloading and quenching can be done mechanically with charging machines. After heat treatment, all castings are Brinell tested.

Every melt is made under laboratory control and test bars are cast from each heat. All prototype castings undergo radiographical inspection to ensure that the correct foundry procedure is being employed. Frequent laboratory checks are made of all mould and core sands.

The centrifugal process

Appreciating the possibility of the centrifugal or "spun cast" process when correctly employed for suitable types of work, the company have made a special study of this method of casting. In addition to the improved

Fig. 7. Two cast brackets produced from a conventional sand mould.

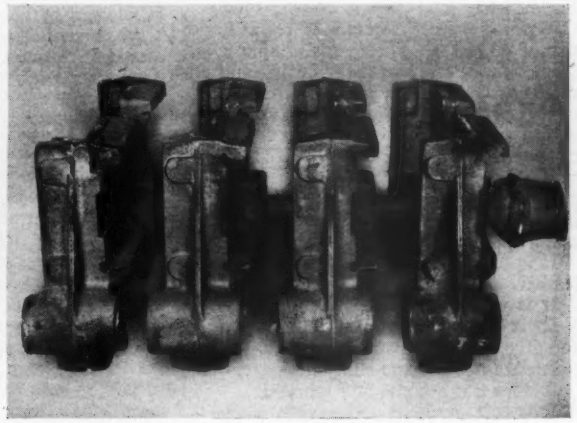


Fig. 8. Eight cast brackets produced by centrifugal casting.

physical qualities, quality of finish and dimensional accuracy associated with the process, certain other important advantages have become evident. In particular, because of the higher yield, it increases the foundry output from the metal in the ladle, and, also, gives better satisfaction to the customer by improving the ratio of the rough casting-to-finish machined casting weight. In one case, the yield was increased from 50 per cent to 70·7 per cent by changing from static cast to centrifugal casting. Naturally, such improvements depend on the suitability of the job for production by the centrifugal process, and it is not correct to assume that these results can be obtained with every type of casting.

In another instance, a casting previously produced from sand moulds and weighing 21 lb as despatched to the machine shop, was reduced to 19 lb by changing over to the centrifugal process. This was obtained by giving smaller machining allowances, and by the absence of the taper cores, etc., necessary to feed bosses successfully when using the static method of casting.

Spun cast components

Some examples of centrifugally cast components are provided by the accompanying illustrations. For example, Fig. 3 shows a cluster of 20 steel body mounting brackets, each weighing approximately 14 lb. The mould, comprising an assembly of "blown" cores, is rotated at 180 r.p.m., the metal being run from a central stalk. Fig. 4 shows 16 rear spring front bracket castings, each weighing 18 lb and produced in a similar manner.

The spring brackets in Fig. 6 weigh 100 lb each when delivered to the machine shop. Originally, they were static cast singly (Fig. 5), that is, in sand moulds, and weighed 200 lb each complete with runners, etc.; in contrast, after changing over to the centrifugal process (Fig. 6) they weighed only 143 lb complete with the centre stalk. Thus, by the centrifugal process a 77 per cent yield per ladle is obtained. Because of the absence of risers, ramming up of the top mould is considerably simplified and moulding costs slightly reduced. When pouring, the metal is run into the stationary mould from a remote runner, which is then immediately frozen from the top and the machine set into rotation. In all the other examples given here, however, the metal is poured while the mould is rotating.

Fig. 7 shows a static cast rear spring rear bracket, produced in pairs weighing a total of 115 lb: note the considerable amount of "waste" metal which must be removed by fettling. The dressed weight of 46 lb, that is, 23 lb each, gives a yield of 40 per cent. In contrast, by changing to the centrifugal process, the brackets were produced in stacks of eight castings (Fig. 8) weighing a total of 236 lb, this representing a yield of 71·2 per cent. In addition, the dressed weight is now 21 lb per casting, 2 lb being saved by reducing the machining allowances and by improved detail.

Fig. 9. A stack of 32 spun cast buffer brackets. They weigh 202 lb and represent a yield of 63.4 per cent.

The buffer brackets in Fig. 9 have always been spun cast, and thus comparison cannot be made with the static process. They are produced in stacks comprising 32 brackets and weighing 202 lb. When dressed, the castings weigh 4 lb each, i.e., 128 lb per 32, this representing a yield of 63·4 per cent.

Inspection jigs

A very thorough inspection system ensures that every casting leaving the foundry is dimensionally correct. When the quantities are sufficiently large, this includes the use of jigs designed specially to suit each particular casting, these incorporating gauging points which enable instant detection of incorrect dimensions, misshapen, or other surface faults.

Whilst in the jig, pre-determined location "spots" are machined, these comprising three small circular spots and two dowel holes corresponding to the position of the location points provided on the fixtures used in the machine shop. These are produced by drilling and spot facing through hardened steel bushes built into the jigs.

This system possesses several important advantages. In particular, it ensures that expensive machining time is not wasted on castings which might not "clean up" at a later stage. Also, setting-up in the machine shop is considerably simplified. Another minor, but often important point is the fact that it eliminates dispute between the foundry and machine shop regarding the responsibility for spoiled work.

Chamois Leather

A NEW substitute for chamois leather is marketed by Walter Shaw, Ltd., Marylebone Mews, New Cavendish Street, London, W.1. It is manufactured in three sizes, 18in × 16in, 24in × 18in and 24in × 21in, and retails at 2s 6d, 3s 9d, and 4s 9d respectively. Its basic materials are Latex rubber and cotton fibres. It is stated to be hard wearing and durable, of standard thickness throughout, and contains no thin patches. The following are the chief claims for the new material:—

It is one-third the price of chamois leather.

It is very much stronger.

It is difficult to tear, even when roughly used.

It does not become slimy in water.

It does not become stiff when dry.

It is soft, resilient and pliable when wet.

It has an excellent abrasion resistance.

It has a longer life than leather and improves with use.

It can be washed out in warm water with any soapless detergent.

If allowed to become very grimy it can be boiled (boiling time not to exceed two minutes). (1911)

Heat Transmission

AN international discussion is to be held in London from September 11th to 13th, 1951, on the subject of post-war heat transfer development. Participation by British and European engineers is being co-ordinated by the Institution of Mechanical Engineers, while the American Society of Mechanical Engineers is undertaking the administrative details. Preliminary plans indicate that the following subjects will be among those discussed:— Recent information on physical properties of gases and gas mixtures; Measurement problems related to heat transfer; Mass transfer and heat transfer; Heat transfer by sweat and film cooling; Condensation at high speed flow; Heat transfer to liquid metals, and The analogy method for the solution of conduction problems. (1905)

Aluminium Gears

MADE of aluminium alloy bonded to a steel hub with a molecular process originally developed by Al-Fin, a new gear has been developed by the Al-Fin Division of the Fairchild Engine and Airplane Corporation, Hagerstown, Maryland, U.S.A. It is claimed to be lighter and stronger than moulded resin and fibre gears and will withstand heavier loads. It is also claimed that, since the gear is hobbed and shaved, it is as quiet in operation as the composition type. It has a tensile strength of about 15,000lb per sq in, and has withstood shear tests of the bond up to 98,000lb. A Brinell hardness of 85 to 120 can be obtained, depending upon requirements. (1915)

DIRECTIONAL STABILITY

A Study of the Factors Involved in Private Car Design

By G. E. Lind Walker, M.A., A.F.R.Ae.S., A.I.Mech.E.

A survey of the present state of knowledge on the subject of the directional stability of cars shows it to provide a sound basis upon which the conditions for full course stability can be defined. From this, the actual design procedure for ensuring that a car should show optimum course stability has been developed. The two factors controlling the stability of the car upon its wheels are generally known. The first and predominant one is the flexibility of the tyres, which yield whilst supporting a side load. Secondly the geometry or even the elasticity of the suspension system may sometimes exercise a significant influence.

Before making an investigation of the subject, it is desirable to have some single criterion of the stability of the car upon its wheels. This is provided by the conception of a "stability margin" between the centre of gravity of the car and the "neutral steer line" which is the resultant of all the suspension factors. At any point along the neutral steer line a lateral force may be applied without producing any tendency for the car to rotate about a vertical axis. The car will therefore be directionally stable upon its wheels when the centre of gravity lies in front of the neutral steer line. It will be obvious, however, that changes in the loading of the car will move this line. Recommended values for the stability margins have been found.

Any trend towards lighter cars with smaller engines increases the importance of aerodynamic forces since their magnitude can no longer be reduced in proportion to the weight.

Special consideration must be given to the shape of the body if the drag is to be reduced without impairing the directional stability. A correctly proportioned body will produce a very small turning moment towards a side wind so that the car will of its own volition tend to assume a slightly oblique attitude thus accommodating the lateral slip of the tyres to the applied side force without departing from its original course. A car which fulfils these conditions will show very little response even to a wind blowing in powerful gusts from the side. It is "Course Stable." The demands of straight course stability also have a favourable influence upon the cornering capabilities of the car. An aerodynamically unstable car may become suddenly uncontrollable in fast bends.

A CONSIDERABLE number of works have been published relating to investigations into the directional stability of cars, but individually they do not provide a sufficient basis for design work. The following notes summarize the findings of a design investigation into the method of securing stability in a new design. In some cases new work was introduced in order to fill gaps which appeared to exist between the existing works of the various recognized authorities. The project under consideration was a small and ultra-light car and it was early realized that the ratio of frontal area to weight in such a car is far greater than in a normal or larger sized car, since both bodies must be laid out to accommodate two persons abreast. Air resistance will thus form a greater proportion of the total resistance for the smaller car than for the larger type, even though the latter may be faster.

Some of the recent French ultra-light cars indicate that this is also the opinion of their designers, since they have been most carefully streamlined, even though not intended for high speed use. Pursuit of low drag body forms led to the realization that wind forces also affect the directional stability, and here again the small car, in spite of its lower speed, might be more seriously affected than the larger types. The relatively greater ratio of laden/unladen weights may also make the small car less stable upon its tyres than the larger car under some conditions of loading. The realization of the increased importance of the two factors of weight distribution and air pressure in the case of the smaller car instigated the detailed investigation of the whole matter.

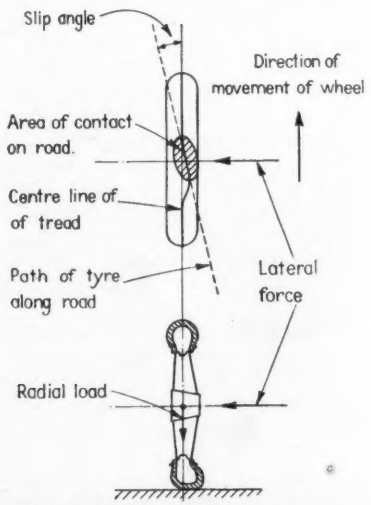


Fig. 1. Deformation of a flexible tyre under the combined effect of radial and lateral forces.

Any car, whether stable or not, once established on a level surface, will continue to run steadily on its course until it is disturbed by one or other of two influences:—

(a) Lateral gravity forces, produced by inclinations of the surface.

(b) Aerodynamic forces produced by a wind inclined to the path of travel of the car.

The subsequent behaviour of the car will vary according to whether it is stable or unstable. The course-stable car will try to regain its original course whilst the unstable car will proceed to turn farther away from it. This applies to a curved course even more than to a straight course. Consideration has been given to the alternatives of front or rear wheel drive, but rear wheel steering has been given no consideration in view of the adequate demonstration quoted by Olley (1947) of its inherent instability.

Ground stability

The pneumatic tyre in universal use on motor cars is flexible not only radially to absorb road undulations, but also laterally. Its characteristics

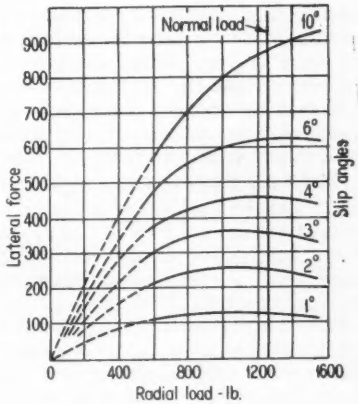


Fig. 2. Tyre slip angles.

Tests of a 7-16/4 tyre on rim 4.50 E at 30 lb/sq. in. inflation pressure running against 67in. diameter steel drum.

DIRECTIONAL STABILITY (Continued)

have been described by several workers, perhaps first by Bradley and Allen, and then by Evans, and more recently by Bull, whose works have supplied data on the more diverse characteristics of the tyre. The car designer should now expect to obtain all the requisite data relating to the tyres he intends to use from their manufacturer, although tyres of the same size may vary slightly in some of their characteristics when they come from different manufacturers. The availability of this data is most important because of the great influence that the tyres exert over the directional characteristics of the car. Fig. 1 illustrates the effect produced by a lateral force acting upon a pneumatic tyre as it rolls along.

The small area of tread in contact with the road is pulled to the side as the wheel rolls. When this occurs, a point on the tread approaches the road in its undeflected position, then moves transversely relative to the rim as it takes the load, and the wheel actually traces a path inclined by the "slip angle" to its own plane. The slip angle of the tyre (a misleading but well established term) depends upon the load carried by the tyre, the axial thrust, the inflation pressure and the camber at which the wheel is running inclined to the road surface, besides several factors in the design of the tyre itself and of the wheel.

Fig. 2 is a type of diagram introduced by Olley to show the relationship between slip angle, lateral force and the radial load on the tyre. It appears that the slip angle may also vary with the driving or braking thrust transmitted by the tyre. This change is very small and from the inconsistency of the available data its nature would seem to vary with different makes of tyre. When a lateral force is applied to the simple vehicle having flexible tyres but no suspension system, and initially running on a straight course, it will no longer continue to move along its own axis. In this case, the front and rear wheels will each move along tracks that deviate from the vehicle axis by their own respective slip angles.

If the front and rear slip angles are not equal the vehicle will move on a curved path, i.e., it will "yaw," a term that is used to signify angular movement of the vehicle about a vertical axis through its centre. As has been explained by Olley, if due to a force applied at the centre of gravity the rear slip angle is greater than the front, then the car will run on a path of increasing curvature. It will turn towards the disturbing force, to which will be added the centrifugal force,

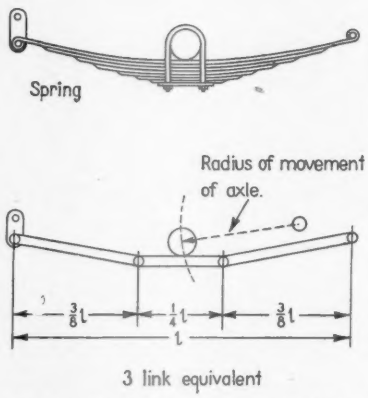


Fig. 3. Three-link equivalent of leaf spring.

due to the turn, and such a vehicle is said to "oversteer" and is unstable.

A stable vehicle has the front slip angle greater than the rear, so that the centrifugal force opposes the initial disturbance, and the vehicle tries to run straight. This is the condition of "understeer." The initial disturbing force may be either aerodynamic, acting at the centre of pressure, or the mass of the vehicle acting at the centre of gravity as the result of either centrifugal forces or of inclination of the surface upon which it is running.

The spring suspension system of a car complicates matters somewhat, for the lateral disturbing force may also produce angular deflection of the wheels. This fact has long been

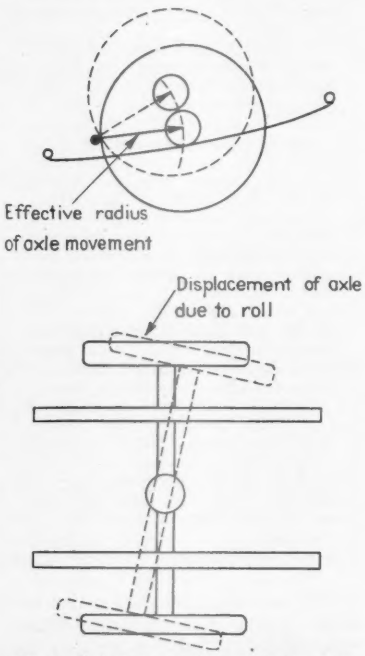


Fig. 4. Roll steer of a beam axle located by leaf springs.

realized. Olley (1937) has described how a leaf spring, controlling the location of an axle, may be reduced to an equivalent 3-link mechanism which gives the same movement of the axle, see Fig. 3. Bastow has shown how, if the springs are inclined, rolling of the car sideways upon the springs will produce steering of the complete axle beam, see Fig. 4.

This may be either an understeering or an oversteering tendency according to which way the springs slope. It is an understeering condition when the springs slope downwards to the centre of the car, and it is particularly useful because it accommodates changes of loading. Bastow also drew attention to the fact that distortion of the suspension linkages due to lateral forces may produce a steer effect quite independent of the angle of roll of the car on the springs. Fig. 5 shows how this may happen in the case of a Dubonnet type of suspension mechanism.

The swinging arm mechanism will probably produce some steering effect as the wheel is displaced if the axis about which it swings is not both horizontal and perpendicular to the centre line of the car. Fig. 6 shows how, if the axis is inclined upwards to raise the roll centre of the suspension system, the wheel will also yaw as it is deflected, since the plane of the wheel remains aligned on a fixed point along the axis whilst the wheel itself moves bodily outwards, in this case producing understeer if the arms lead towards the centre of the car. The directional changes of the wheel in this case may be considerable.

There is similarly a considerable change of wheel direction in the common types of swing half-shaft rear suspension, where the wheel swings about an inclined axis passing near the centre of the axle and out to a wheel radius arm pivot on the side of the car. This arrangement may produce an appreciable understeer effect although the two wheels both turn towards the centre of the car, because the outer, more heavily loaded, wheel tends to take directional control of the rear of the car. The small change of camber angle of the rear tyres with this system may also be favourable to stability.

Perhaps the most simple method of producing a slight "roll understeer" is to arrange the steering linkages so that the front wheels toe-out very slightly as the springs are compressed. When the car rolls both wheels will turn outwards from the bend, thus providing a definite and easily varied amount of understeer.

In addition to yaw of the wheels

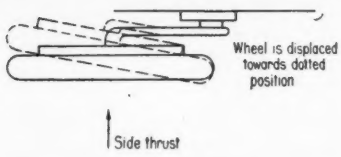


Fig. 5. Steer produced by suspension distortion.

Bending of arm in Dubonnet-type suspension. This may increase or decrease stability according to direction of arm.

produced by the characteristics of the suspension system, the lateral flexibility of the tyres also varies with the camber of the wheels. With most types of independent suspension, as the body of the car rolls, the wheels also lean, or change their camber angle, in the same direction. This results in increased slip angle of the tyres for given conditions of loading. It is for this reason that cars with independent front suspension have a natural tendency to be directionally stable.

The various factors just mentioned are all contributary to the final directional characteristic of the car but no single one affords any criterion of what the directional stability will be. It is thus necessary for design calculations to find some other factor that is a unique criterion of directional stability, and which preferably is non-dimensional. One answer to this problem has been given by Stonex. A circling test is made at constant radius on the skid pad, at varying speeds. As the speed varies so also does the steering angle required to hold the course. If the car is stable the steering angle will increase with speed. Stonex recommends an increase of 2-3 deg of steering angle above the dead slow setting when the radial acceleration is 0·4 gravity. In acutal fact the car upon which he made most of the tests he describes, appeared generally satisfactory with a value of only 1 deg increase.

To provide a more convenient process of stability estimation for design work, the conception of a "stability margin" has been borrowed from aeronautical practice. It is the horizontal distance, expressed as a percentage of wheelbase length, between the centre of gravity and the centre of reaction of the road forces, which may be called the "neutral steer line."

The neutral steer line is defined as a line in the fore and aft vertical centreline plane of the vehicle, anywhere along which a lateral force may be applied without producing any yaw of the vehicle as it proceeds along its track, steering wheel fixed (provided that the limit of adhesion of the tyres is not exceeded). If the centre of gravity of the vehicle lies behind the neutral steer line then the vehicle will yaw towards a lateral force applied at the centre of gravity, so that centrifugal force will be added to the disturbing force and the rate of yaw will increase. This vehicle is unstable. To be stable a vehicle must have its neutral steer line located behind the c.g., so that it under-steers.

The neutral steer line of the simple car, with no spring suspension system, is vertical, through the mid point of the wheelbase if the loading is equal on the two axles. Reduction of tyre pressure makes the neutral steer line move away from the softer end. For a car with a spring suspension system and evenly loaded axles, the neutral steer line will pass through the mid point of the roll axis. If the centre of gravity is moved towards either end the neutral steer line will move in the same direction, but generally by a slightly less amount.

The effect of roll steering by the suspension system is to slope the neutral steer line, though it still passes through the same point on the roll axis. An understeering tendency of the suspension slopes the neutral steer line aft at the top, increasing the margin of stability. Conversion of Stonex's figure makes the recommended stability margin between 4 per cent and 6 per cent of wheelbase length, which is of similar order to the values expected for aircraft.

It must be emphasized, however, that the neutral steer line is not fixed. As the loading of the car is varied and the centre of gravity is moved, so also does the neutral steer line move back and forth, though normally to a lesser extent. It is also affected by tyre pressures, reduction of pressure at the front, moving the neutral steer line aft and vice versa. In fact, to be significant the neutral steer line and the stability margin must be quoted together with the applicable conditions of loading and tyre pressures—this again is analogous to aircraft practice where the stability margin varies with the loading and the speed.

Direct experimental measurements of the positon of the neutral steer line of a car have not yet been made but the equipment shown in Fig. 7 would seem to be theoretically correct. The car is driven, steering locked, along a straight course on a flat surface for a distance of perhaps 100 feet. Any angle through which the car yaws is carefully measured. The car is then returned to the start and a horizontal side force is applied to it by means of a cable which passes through pulleys to support a hanging weight mounted on a small rail truck. By this means the load is kept constant. The truck is moved by a towing cable so that it always remains abreast of the car, applying a true side load.

A series of runs are made with the steering locked in the one position whilst the side force is applied at different points on the car. These points are at different fore-and-aft locations and at different heights. By measuring the yaw produced during each run the neutral steer line can be located at each of the different heights. A side plate may conveniently be fitted to the car to facilitate attachment of the cable to the car at the various positions. It would be interesting to compare the results obtained with lateral forces of different magnitude before selecting any value as standard. A figure of 0·25 times car weight which is representative of normal arduous cornering would probably prove a good standard.

A side force of this magnitude will produce considerable lateral slip of the tyres, thus the car will always slip from its course towards the force, but the attitude which it assumes will control its ultimate path, hence only the angle of yaw is required on each run. The disadvantage of this method of testing is that it is restricted to low speeds, whilst both the flexibility and the adhesion of the tyres vary somewhat at high speed, and in some cases it would seem necessary to make allowances for this.

The disposition of the neutral steer line of the car to provide an adequate stability margin under all conditions of loading is the outcome of correct arrangement of the geometry of the suspension system and the selection of tyres to suit the demands of the particular car which is being supported.

The paucity of published results for the stability criteria of motor cars as compared with aircraft is unfortunate, for it leaves room for speculation on many highly esteemed motor

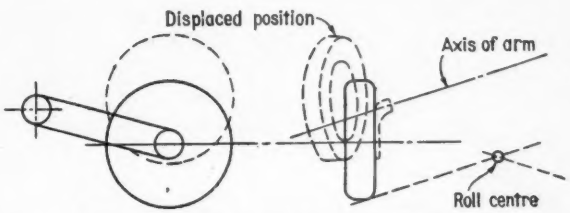


Fig. 6. Swing arm suspension with inclined pivot.

DIRECTIONAL STABILITY (Continued)

cars of the recent past. Analysis of their suspension geometry indicates that in many cases these cars had a negative stability margin, steering locked. They derived their good handling characteristics from the self-aligning action of the front tyres, which with high-geared, reversible steering could follow their own natural, straight track without producing much reaction at the driver's hands. On the other hand the lack of inherent stability of the complete machine assisted it in making rapid changes of direction.

It is certainly preferable for general service that the car should be directionally stable with the steering fixed, particularly since the steering tends to become lower geared and less reversible. But in view of the marked directional sense of the front tyres, there seems little need for a large stability margin which carries with it the penalty of slow response when changes of direction are required. It may be that the value of 4 to 6 per cent of wheelbase length is a maximum rather than an optimum value for the stability margin.

Aerodynamic characteristics

The flow of air over a car body produces a distribution of pressure, the summation of which is a net force on the car: for dimensional clarity this force is resolved into six components:—

(a) Resistance.
(b) Side force.
(c) Yawing moment about the centre of gravity.
(d) Lift.
(e) Pitching moment about the centre of gravity.
(f) Rolling moment about the c.g.

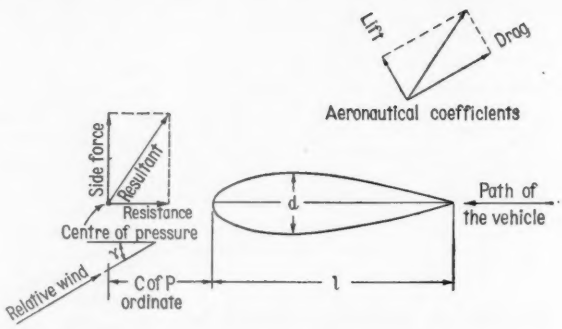


Fig. 8. Disposition of aerodynamic forces acting on body.

In very high speed cars all these components have some effect upon the road adhesion and directional stability. The lift and pitching moment, for instance, may transfer loading from the tyres at one end of the car to the other end, with consequent movement of the neutral steer line.

However, for the purposes of this work only the first three components will be considered. This in fact assumes that the aerodynamic centre of pressure is at the same height as the centre of gravity. Under this limitation, when the relative wind comes from directly in front of the car, resistance will be the only force produced. When the relative wind blows from one side it will produce in addition, side force, and probably yawing moment.

Aerodynamic data for aircraft is based on co-ordinates of wing direction and mutually perpendicular axes. These in fact are the axes of movement of the aircraft. In the case of the motor car it is more convenient to resolve the forces along the centre line of the car and mutually perpendicular axes. The centre line of the car is also, within close limits, its line of travel on a straight course. The term "resistance" is used for the air force opposing motion of the car along its own axis and "side thrust" perpendicular to the direction of motion, pushing the car off the road.

The resistance, side thrust, and yawing moment may be combined vectorially to give a single resultant aerodynamic force, acting horizontally through the centre of pressure as shown on Fig. 8. From this figure it will also be seen that the resistance coefficient:

$$C_R = C_D \cos\gamma - C_L \sin\gamma$$

and the side thrust coefficient

$$C_S = C_L \cos\gamma + C_D \sin\gamma$$

where γ is the angle of attack to the wind. Fig 9 based on tables given by Bairstow, shows the results of measurements made on a streamlined, aeroplane type of body, with fineness ratio $l/d = 6$. The coefficients are in terms of the sectional area of the body and based on the axes used for automobile work, although the aeronautical type of coefficients have been used, so that

$$\text{Resistance} = q.\ A.\ C_R \text{ lb}$$

and similarly the yawing couple $= q.\ A.\ l.\ C\psi$ lb-ft, where:—

q = dynamic head of fluid $= \frac{1}{2} . \rho . v^2$ at normal speeds.

v = relative velocity between car and wind, ft/sec.

ρ = density of air, slugs/cu ft.

A = projected frontal area of car, sq ft.

l = length of body, overall, feet.

It is important to note that the resistance varies very little with the angle of attack, owing to the helpful component of the lift vector. The most significant feature, however, is that for this long, streamlined body, at small angles of attack, the centre of pressure actually lies a considerable distance in front of the nose. A very small side force can therefore produce a considerable disturbance of direction. Tests made by Weiselsberger, shown by dotted lines, indicate that for a similar body, but of rectangular cross section, both coefficients are considerably higher. The resistance coefficient, value about 0.035, is very low compared with the values found for motor car bodies.

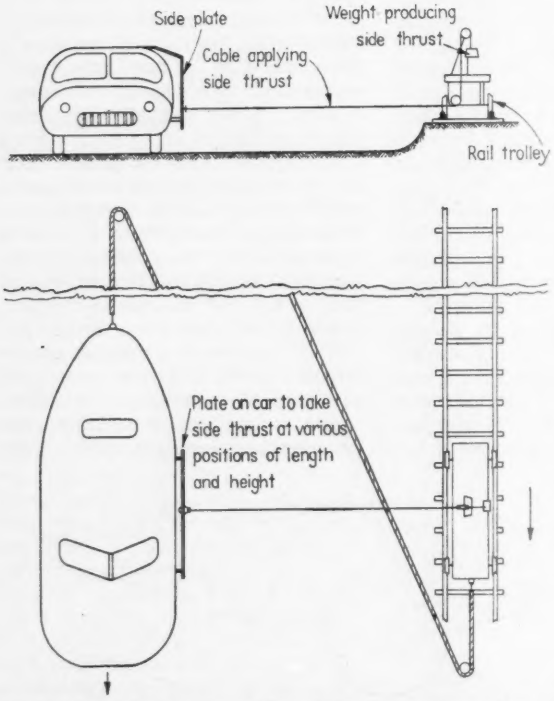

Side plate
Weight producing side thrust
Cable applying side thrust
Rail trolley
Plate on car to take side thrust at various positions of length and height

Fig. 7. Apparatus for determining position of neutral steer line.

Present styles give values of between $C_R = 0.5$ to $C_R = 0.8$, although experiments such as those made by Zeder have shown that values of $C_R = 0.25$ are practicable without the use of elongated tails, and that a value of about $C_R = 0.17$ could be obtained for a touring machine. Much of the great discrepancy between this and the value of 0.035 for the real streamlined body may be attributed to the projecting wheels and a few other essentials, and the remainder to the interference effect of the adjacent ground upon the flow of air round a car.

Large angles of attack

The tests of aircraft bodies were not extended to the large angles of attack which could be reached by motor cars, where a side force coefficient of $C_S = 1 \cdot 0$ might be reached in extreme conditions. In this case the side force acting on the body is considerable,

$$F = \tfrac{1}{2} \cdot \rho \cdot v^2 \cdot A \cdot C_S \text{ lb.}$$

Force is thus proportional to v^2 and since also at small angles of attack air speed and road speed will be very nearly the same, a comparison may be drawn between side force due to wind and the centrifugal force C due to cornering.

$$\text{Centrifugal force} = C = \frac{m \cdot v^2}{r \cdot g} \text{ lb.}$$

F and C would thus be expected to bear a constant relationship to each other. Consideration of a car of frontal area $A = 20$ sq ft and side coefficient $C_S = 1 \cdot 0$ and equating F to C will show the radius of corner which produces a lateral force equal to that due to wind, for any given speed and weight of car. Reference to the cornering tests made by Deitz and Harling then provides the practical limit of road adhesion. Table I indicates the results which were obtained.

TABLE I

Total weight of car, lb.	Equivalent corner radius, yards	Practical limit of speed, m.p.h.
3,000	1,260	110
2,000	850	95
1,000	420	75

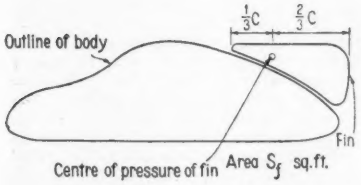


Fig. 10. Location of centre of pressure of fin.

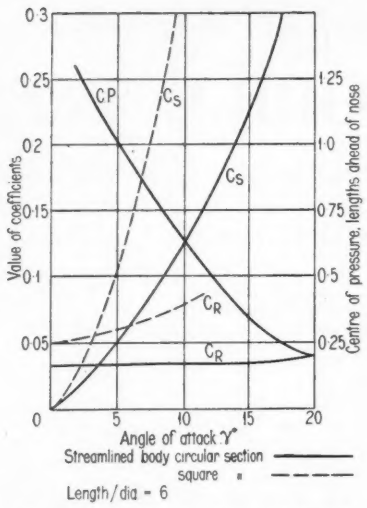


Fig. 9. Aerodynamic characteristics of streamlined bodies.

This shows that the ultra light car could be blown bodily off the road at a relative wind speed of 75 miles per hour, whilst a car of normal weight might be controllable up to about 110 miles per hour. The significance of this analysis lies rather in the illustration it provides of the significance of aerodynamic forces upon the ultra-light car at speeds which are already being reached in touring, than in any actual value the figures may have. The effect of a disturbance of this nature will be greatly magnified if the force takes effect far to the front of the body.

When the ultra-light car is streamlined, care must be taken at the same time to ensure that the centre of pressure is restrained in a rearward position. The fineness ratio l/d will probably be less than 6/1 for a car body, and also its natural shape will tend to retain the centre of pressure relatively near the nose. In fact, for a non-streamlined body the centre of pressure will probably be quite near the middle. As is shown by Fig. 9, the centre of pressure is not generally stationary, and it will tend to move aft with increase of the angle of attack of the car on the wind.

The centre of pressure of a streamlined body may be moved aft by the addition of fins at the rear. Whilst at small angles of attack the centre of pressure of the ideally shaped body, $l/d = 6$, is some $1 \cdot 3$ lengths in front of the nose, for more practical shapes it lies only about $0 \cdot 42$ lengths in front of the nose for l/d ratios between 10 and 3. As the body becomes more deformed, into the motor car, the centre of pressure moves further back, being about $0 \cdot 1$ lengths behind the nose of a production streamlined saloon, and near the middle of the car if the form is not streamlined. In this last case there is no problem of aerodynamic instability.

The actual side force acting on a body of practical streamlined form, under various conditions may be expressed as

$$F_b = (0 \cdot 014 + 0 \cdot 008 \; l/d) \cdot A \cdot q \cdot \gamma$$

at any angle of attack γ degrees and will take effect at the centre of pressure for the bare body form.

Addition of fins

Fins can be added at the rear of the body to bring the overall centre of pressure aft to any desired location. Fig. 10 illustrates such a fin, showing its centre of pressure to be about ⅓ of its length back from the leading edge. The side force produced by the fin is given by the formula.

$$F_f = 0 \cdot 045 \cdot S_f \cdot q \cdot \gamma$$

where S_f is the projected side area of the fin in square feet.

Now the total resultant side force acting on the car is

$$F = F_b + F_f$$

and the point at which it takes effect can be determined, as shown in Fig. 11, by taking moments about the nose of the body.

$$F \cdot x_{c.p.} = F_f \cdot x_f - F_b \cdot x_b$$

$$\therefore \quad F_f = \frac{1}{x_f} (F \cdot x_{c.p.} + F_b \cdot x_b)$$

from which the required area of the fin at a given position can be determined to produce any requisite value of $x_{c.p.}$. It should, however, be realized that these empirical equations are liable to considerable variation with different shaped bodies, thus the overall centre of pressure will only be certainly positioned with the aid of wind tunnel testing When this has been done the centre of pressure should be arranged to remain always between close limits in the required position.

(To be continued.)

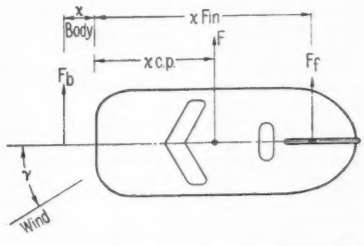


Fig. 11. Analysis of side forces to find the centre of pressure of complete car with stabilizing fins.

Institution of Mechanical Engineers, Automobile Division

AUTOMOBILE TEST RIGS

A Description of Typical Equipment, Indicating its Scope and Results

By L. H. Dawtrey, M.I.Mech.E.*

(*Continued from page 265*)

A SIMILAR rig is shown in Fig. 17, where a variable-speed electric motor drives through a "big car" gearbox intended to give a further speed range. In fact, this feature could not be used because the drive is reversed to normal, giving unsuitable lubrication for the plain bushes of the layshaft of this box. However, the box is retained because of the plate clutch it incorporates. A heavy 3/1 ratio gearbox serves to raise the output speed to the water brake, while the car gearbox on test is provided with water cooling and a heat exchanger. Fig. 18 illustrates a car gearbox rigged for overrun tests. Here an engine unit provides the only resistance, as it does in the vehicle when overdriving, with the engine acting as a brake.

Synchromesh wear tests

Life tests for synchromesh engagement of gears have been carried out on a comparative basis as shown in Fig. 19. Here the change-speed lever is operated automatically from top to second gear, and *vice versa*. The work which synchromesh cones have to perform is to change the speed of the engine clutch plate, and the heavier the plate the more severe is the wear and tear on the synchronizing parts. Therefore, for accelerated wear tests, the clutch is weighted up to double normal weight.

The gearbox output shaft is driven at a speed equivalent to 40 m.p.h. on the road. A two-way air jack is used to operate the change-speed lever, using some old aircraft equipment, namely, a Dunlop air-wheel brake differential steering unit, as a control valve. This allows controlled rate of operation of the change-speed lever. The drive to the control valve is by linkage from an 8 r.p.m., geared electric motor, thus giving eight gear "changes" per minute. A "life" before the onset of "crashing" gears under these severe conditions is around 30,000 gear changes.

Rear Axle Test House and Chassis Dynamometer. The test house illustrated in Fig. 20 was primarily designed for rear axle testing, but accommodation was provided for a complete vehicle to be rigged with its rear axle jacked up into line with the dynamometer. Air-cooling for the exhaust system is necessary.

When this test house is adapted to take a vehicle, the sound-proofed step-covers used as propeller shaft guards are removed to stores, leaving ample space for the vehicle, which can be power-tested, or run at critical boom or vibration speeds or other suitable conditions.

Experience has shown that noisy or quiet axles can be determined with certainty on this test, using the petrol engine drive and dynamometer flywheels only. An engine throttle-lever located adjacent to the axle allows the operator to reproduce road driving conditions. No energy is absorbed by the dynamometers, but only by the flywheels which are proportioned to equal the weight of a saloon motor car. The test house is not intended for mass-production rates of test, but four or five axles can be tested per hour, if necessary.

The layout is a large sound-proofed test house with dynamometer rooms at each side, while at the far end is the engine drive room, which is also used for axle stripping and investigation work.

Fig. 21 shows one of the dynamometer brakes and the attached flywheel. The amount of flywheel mass can be adjusted. A car petrol engine is used for the drive because of its flexibility. This unit is mounted at each end on ball bearings on the crankshaft axis, the whole mass being balanced. Thus the torque reaction of the engine, instead of being absorbed by fixed engine mountings, is supported by a torque arm and test-brake spring-balance. By these means the power input can be measured.

All external engine connections must be very free so as not to affect torque readings The most difficult one, that of the exhaust pipe, is overcome by turning up the silencer outlet pipe into a large diameter stack-pipe. This gives an extractor effect, and allows exhaust gases to be conveyed away without metallic contact and quite without fumes. Water cooling is provided for the car gearbox that drives the propeller shafts conveying power to the axle, as shown in Fig. 22.

Air flow

Flow Rig. This is a useful rig tending to be more widely used in the modern experimental shop. Its purpose is to measure air flow, particularly for inlet and exhaust passages of poppet valve engines. One is illustrated in Fig. 23.

It is most valuable in the development of ports and manifolds. By this technique the port area may be made smaller than the valve throat area without loss of gas flow. This leads to better water spaces around the valve ports, and brings the stem guide as close as practicable to the valve head, while retaining full guide cooling. Sometimes the absolute maximum air flow is the object.

The rig illustrated consists of an air intake drum carrying the sharp-edged circular orifice through which the air flow is measured. Beneath the table is a Reavell compressor operated as an exhauster or suction pump. This has a suction capacity of 200 cu ft per min at 2,000 r.p.m. at a pressure difference of 10 inches of water.

In the pipe to the exhauster is a combination of two coupled throttle valves, one giving variable opening to atmosphere while the other throttles the measured flow. The pump runs at full capacity, but this throttle enables the required flow through the port to be obtained, corresponding to the desired pressure drop across the port, measured on an inclined-tube manometer.

A pulsation-damping tank, which also provides a mounting face, is mounted between the port under test and the suction pump. In the illustration a cylinder head is mounted to the drum for exhaust flow measurements. So far as valve porting is concerned, best flow is achieved with ports with least turbulent flow, that is, those which approximate to a venturi. This means that flow can often be increased by reducing the cross-sectional area at points which, in the original design, had sections suddenly increasing in area (see Fig. 24).

Ports can thus be smaller than is conventional for the valve diameter, and in this one can allow for the fact that the valve occupies very little time at maximum lift compared with the whole period of opening.

Wooden models are often used, jointed in sufficient planes to come apart for shaping ports with Plasticine, and making plaster casts, which are helpful also to the pattern maker in reproducing the port core.

When flow-testing suitable valve ports a length of tube, simulating the cylinder of the engine, is provided.

Damper settings

Shock Absorber or Suspension Damper "Carding" Rig. It would be of real benefit to the automobile industry if the shock absorber manufacturers would standardize the "setting" test procedure. Then designed test machines would be worth while and welcome†.

In the absence of such a machine, a rig shown in Fig. 25, was contrived by assembling various tractor and motor-car parts in production in the factory. A variable-speed belt drive is arranged to operate a combination of farm tractor gearbox and car gearbox, devised to provide speeds suitable to the various makes of damper used, from, in one case 9 r.p.m. for slow-speed bleed setting to 104 r.p.m. for high-speed blow-off setting, at an arm stroke of 6 inches.

A rig or machine of this type is essential, as it is impracticable to return dampers to makers for resetting during the process of development of the vehicle "springing" or "ride," when numerous settings may have to be tried before the best is determined.

The card, or indicator diagram, is the energy picture of the shock absorber, scaled to read arm torque in pound-inch at the damper spindle.

A torsion bar with anti-friction bearings for the linkage measures the spindle torque, but the indicating arm is undamped. Flywheels are used to steady the arm-driving mechanism. Tractor hubs are used for fulcrum points. The rig is also arranged for endurance testing.

Clutch Release Bearing. These bearings are specified to withstand 10,000 operations of declutching without appreciable

* Technical Engineer, Standard Motor Co., Ltd., Chairman of Coventry A.D. Centre.

† Since this paragraph was written, the author has learned that the manufacturers are, in fact, proceeding with the standardization of such a machine.

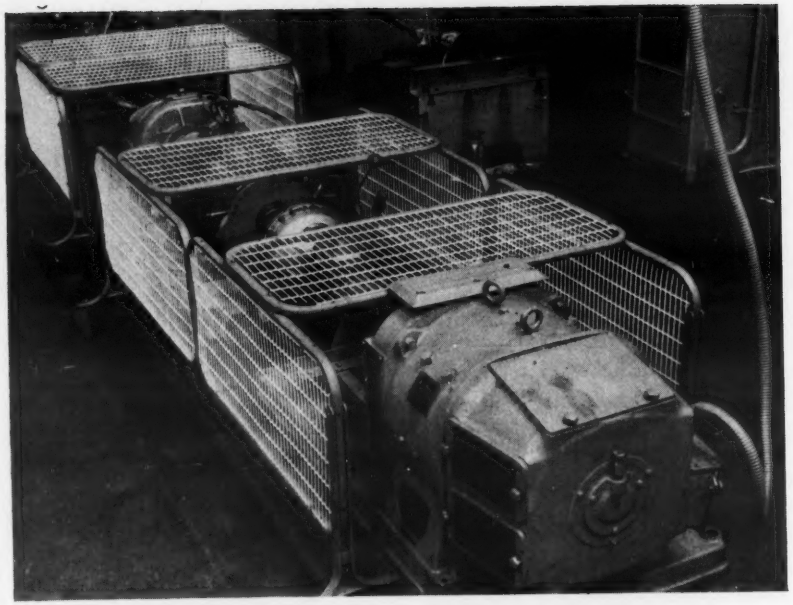


Fig. 17. Gearbox test rig, with electric motor drive, step-up gear, and water-brake.

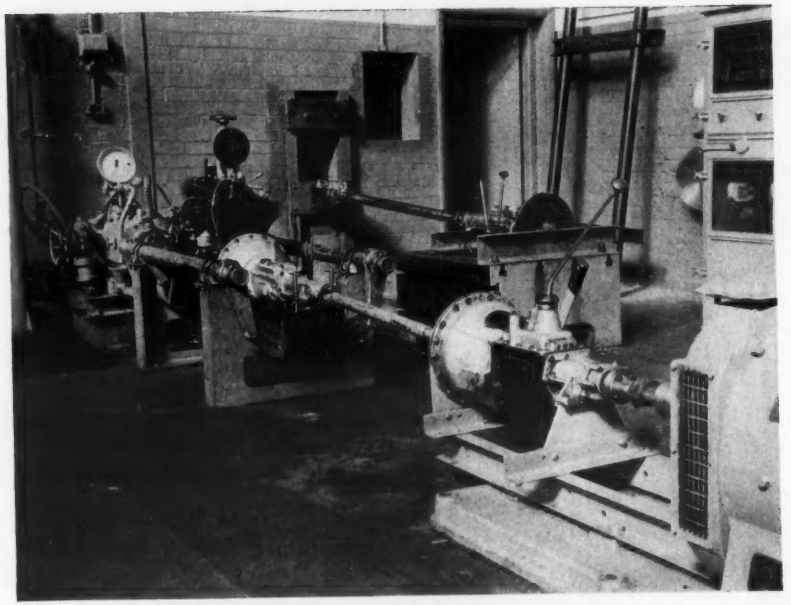


Fig. 18. Gearbox on overdrive test. This uses electric motor drive and an engine as driven unit.

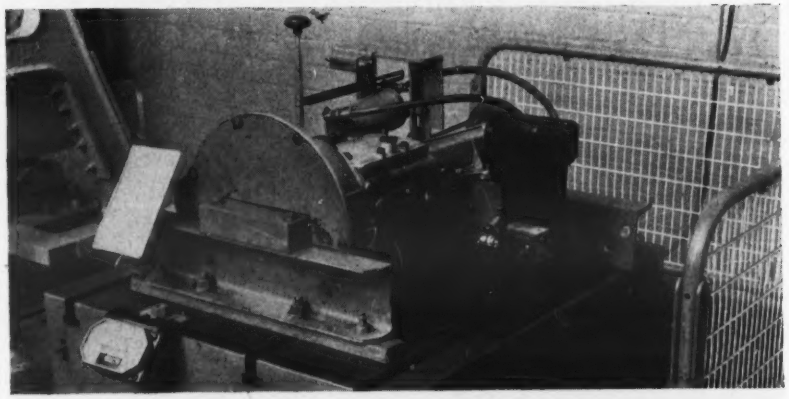


Fig. 19. Automatic test rig for synchromesh gearing.

Fig. 20. Axle or complete car test-house.

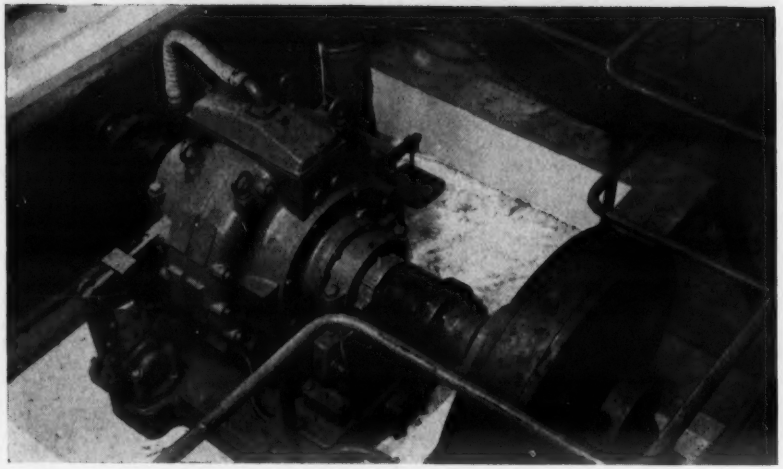


Fig. 21. Axle test-house dynamometer and flywheel.
Two of these units are used, one at each side of test house.

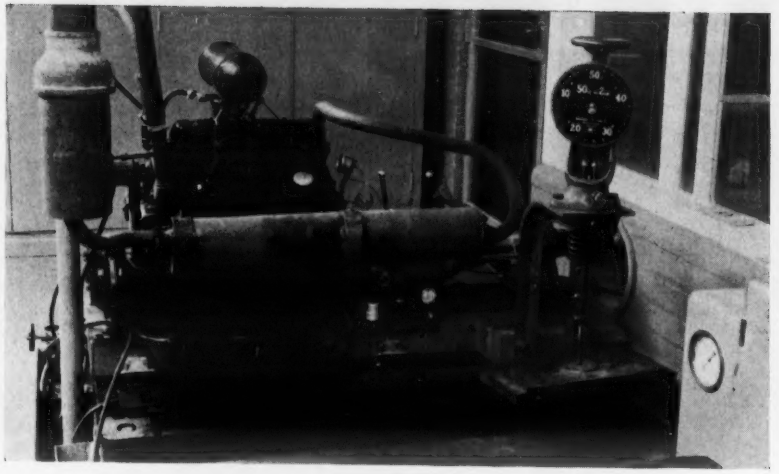


Fig. 22. Engine driving-unit in axle test house.
Adapted for measurement of torque reaction.

Fig. 23. Flow rig. Cylinder head mounted for valve-port flow tests.

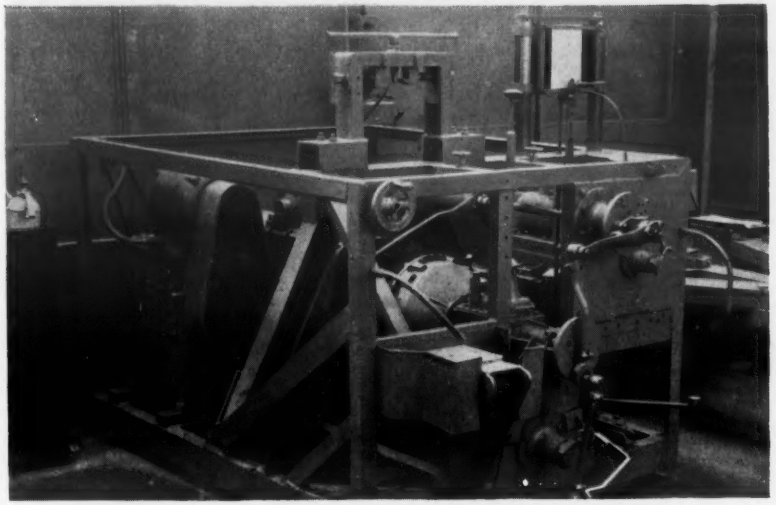
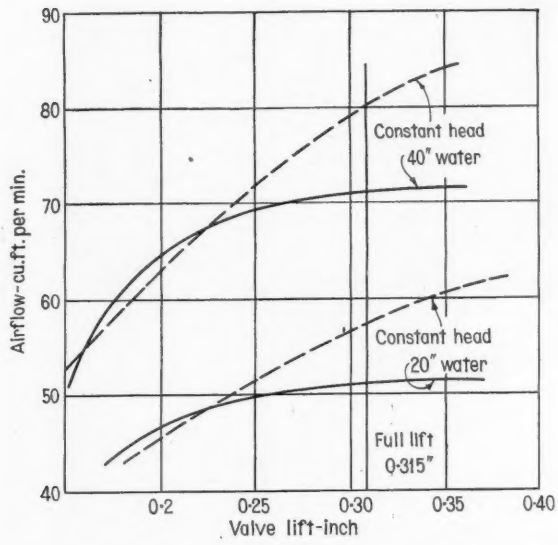
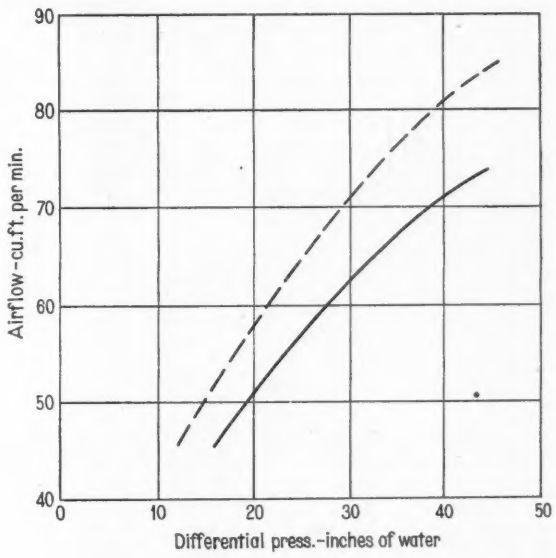


Fig. 25. Shock absorber carding rig.



a Flow curves for 1in diameter valve ports.
———— normal design.
- - - - streamlined development.



b Port flow graph, full lift (0·315in).
———— as designed.
- - - - as developed.

Fig. 24. Flow through valve ports.

loss of grease. Experience shows it to be essential that porous bronze cages are used for the ball bearings.

Fig. 26, illustrates an automatic rig in which an electric motor belt-drives a scrap farm tractor engine at 2,750 crankshaft r.p.m There are no connecting rods or pistons in this unit. An aircraft air jack operates the clutch-withdrawal lever, and is controlled by a small geared electric motor running at 16 r.p.m., thus declutching sixteen times per minute. This rig has proved very satisfactory, for both ball bearing thrusts and graphite types.

Steering Ball Joint. The wear test rig shown in Fig. 27, has a geared electric motor operating steering track rods against the inertia of an oscillating flywheel. The turntables are used to introduce an angular component of motion in certain tests.

Rotating Beam Fatigue Tests. Many engineers use the rotating cantilever test for axle shafts. The author uses an old wood-turning lathe adapted for this purpose. The headstock of this lathe would itself fail under this test and it is, therefore, replaced by a substantial drive in the form of a grinding head capable of withstanding heavy bending moments. The weight carrier is below the lathe bed, and hangs from the " differential or inner end " of the axle shaft.

The use of the machine is not, of course, confined to axle shafts. For instance, although the front stub axle is not usually designed on automobiles as a rotating part, yet stress concentration effects in the design can be determined by a rotating bend fatigue test.

It is, however, often found convenient to run these tests as a beam test, using a pair of axle shafts, secured together with the hub ends as the centre of the bearing cage as shown in Fig. 28. Hanging a heavy weight through bearings each side of hubs, bends the shafts appreciably. A strong coil spring is interposed to support the weight and make the load mass hang steadily. The amount of weight is adjusted to produce the same bending stress as occurs when the vehicle corners at such a speed that the sideways force is equivalent to half its weight, that is, a "$\frac{1}{2}g$ corner". The shafts are rotated at about 600 r.p.m. This produces the same type of failure as that met in service.

For instance, on one type of vehicle, after many years heavy service, or a shorter time if very heavily overloaded, the integral hub flange became a loose ring on the hub, through fatigue-cracking at the junction of flange and hub. This rig test produces exactly similar results, complete with the " cocoa " associated with fretting corrosion, in a few days' running. It also proves useful for testing the whole assembly of a pair of axle shafts, hubs, and road wheels, to show the relative strength of each in fatigue. A further important fact confirmed is the superiority of shafts having integral hub flanges compared with the tapered and keyed designs.

Tractor Power Take-off. The erection shown in Fig. 29, is obviously a rig insomuch as this test bed was not designed to test complete tractors. It is, however, quite effective, and can be arranged for belt pulley unit drive as shown, or direct power take-off with tractor mounted at right-angles to the position shown.

Taper Roller-bearing Wear. In Fig. 30, is shown a rig developed to test taper roller-bearing wear, particularly that at the thrust ends of the rollers, in conjunction with various hypoid oils.

The subject of this test is a centre casing of a hypoid rear axle. The only parts fitted are a hypoid pinion and its two bearings. These are assembled with preload measured in terms of bearing friction torque by the scale pan as shown. No thrust is produced by the drive splines. Oil is pumped through a filter to a tray so that oil can flow by gravity down through the bearings. In order that no nicety of control should be needed for oil flow, a weir (not shown) was fitted so that excess oil could flow freely back by a separate passage to the gear case. Oil temperature during test is 60–70 deg. C., and running speed of pinion 2,950 r.p.m. Starting with 10 lb.-in. torque preload, this is checked every hour. Owing to wear the preload will fall. The ideal result is to reach a state of equilibrium where no further preload is lost while a substantial proportion of the initial preload is retained.

Fig. 31, shows typical results of this hypoid oil-taper roller-bearing test. Some oils are noticeably better than others, and the best is needed for the first factory filling. In addition to these curves, actual measurements of wear are taken which match

Table I. Wear and Loss of Preload in Taper Roller-bearings Running in Hypoid Oil

Hypoid oil type	Running time, hours	Preload		Loss of preload, lb-in	Wear in bearings on overall width		
		At start of test, cold, lb-in	At end of test, hot, lb-in		Small bearing, inch	Large bearing, inch	Total on both bearings, inch
A ..	6	12	0·5	11·5	0·002	0·002	0·004
A_1 ..	20	9·2	1	8·2	0·002	0·0005	0·0025
B ..	40	9·2	6·4	2·8	0·001	—	0·001
C ..	36	9	4·8	4·2	0·001	0·0005	0·0015
D ..	44	11	3·8	7·2	0·001	0·001	0·002

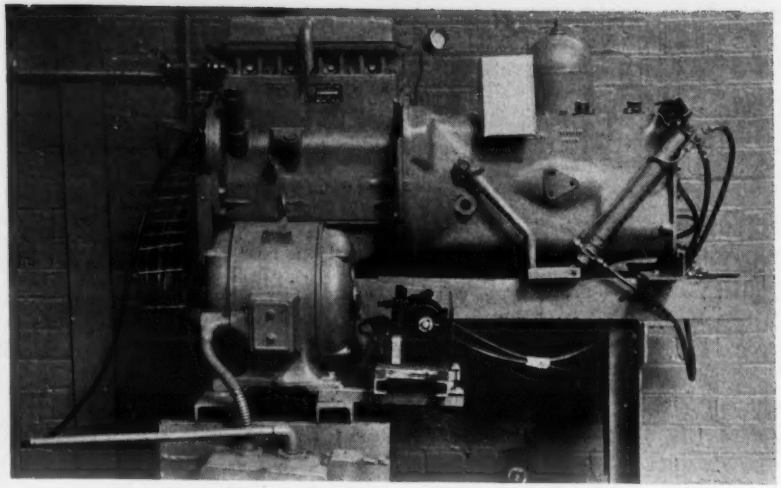


Fig. 26. Clutch release bearing endurance test rig.

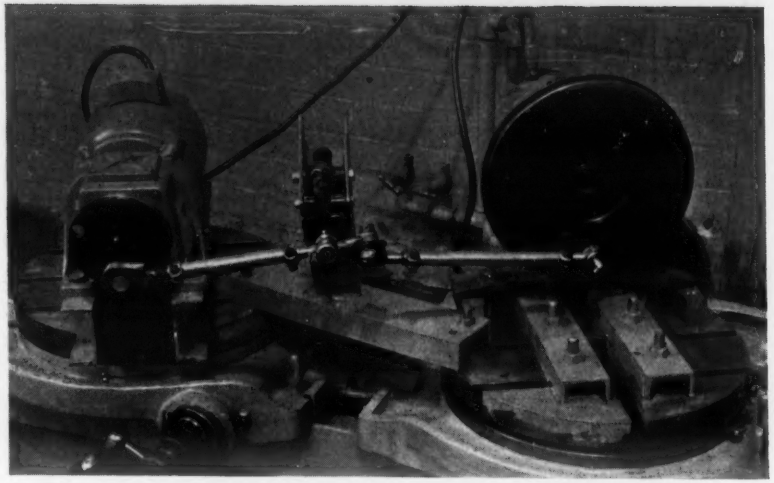


Fig. 27. Steering ball joint wear test.

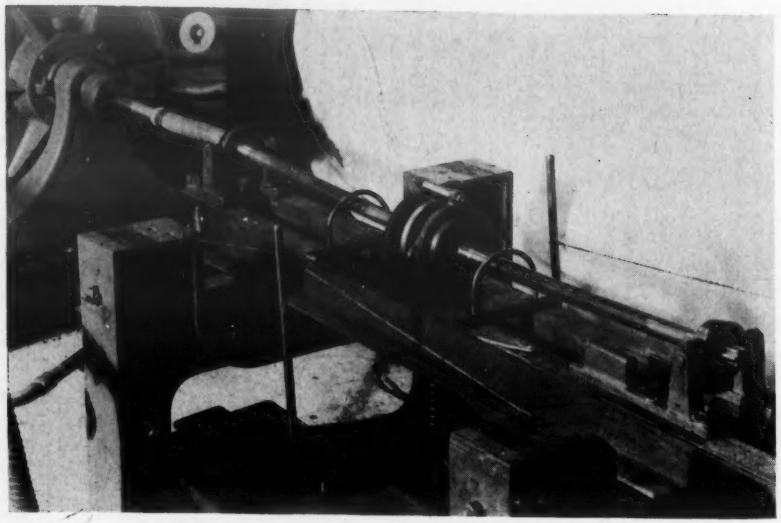


Fig. 28. Axle shaft test rig.

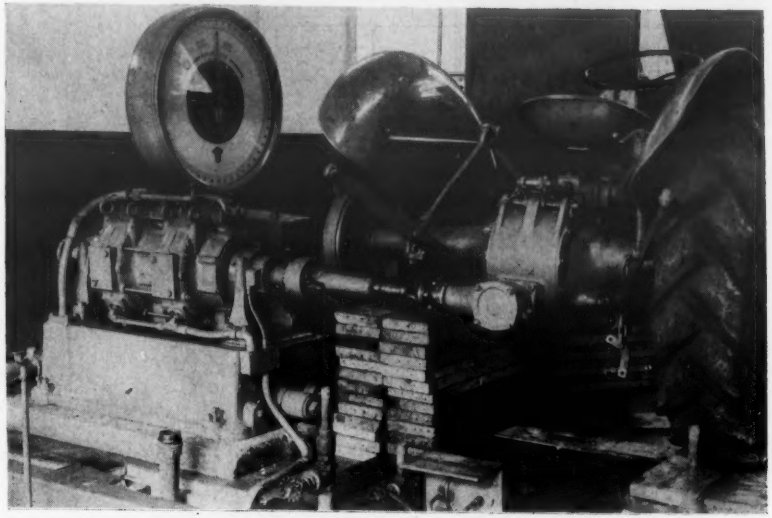


Fig. 29. Rigging complete tractor to engine test-bed.

up with the loss-of-preload curves, as indicated in Table 1. Carefully chosen bearings are needed for strict comparison, however. Observers in two remote establishments have, by these tests, placed differing oils in the same order of quality.

The quality of bearing manufacture is also a factor in the results, the finish of the rollers and tracks being most important.

Hot fatigue of valve steels

Exhaust Valve Breakage. There are indications that the now popular "XB" exhaust valve steel is weak at high temperatures, so that if valve bounce occurs owing to breakage of one of the two springs operating one valve, or if the car is driven too fast at clatter speeds in the lower gears, or if very strong valve springs are fitted, then the valve head may come off the stem with disastrous results in an overhead-valve engine. This denotes hard driving conditions and, while very rare, it has occured. The trouble was unknown in similar engines with austenitic valve steels previously used.

The rig illustrated in Fig. 32, is arranged with two cylinder heads, the top one inverted over the lower one fitted to the engine block in a dark room. The space between the two cylinder heads carries the usual valve rockers, but these are adapted to operate only the valves in the upper head. By these means valves can be broken without the engine damage which would occur if running engines were used for the tests.

No breakage occurs without applied heat. To produce failures a gas torch heats the poppet valve to 750 deg. C. as measured by optical pyrometer. The stem can be seen through the exhaust port and cools off rapidly towards the valve guide. Breakages are produced identical with that which has occurred in an engine.

The test arrangements are to fit only the inner valve spring, which has a fitted load of only 28 lb., and to run the camshaft at 100 r.p.m. faster than the inception speed of valve clatter. This is a speed of 2,000/(4,000 crankshaft r.p.m.). Under these conditions an average valve life, before the head comes off, at 750 deg. C. is 1 hour only, using XB steel.

The failure is caused by hot fatigue and occurs at the smallest diameter subjected to tensile shock, which is, at the same time, red hot. This is, of course, the junction of the fillet radius with the stem diameter. Thus, a larger stem diameter gives a longer test life.

Tests are being made to measure the hot fatigue limits of various valve steels, in conjunction with valve bounce tests.

Calculation of vehicle performance

Since his youth the author has been interested in the accurate calculation of motor vehicle acceleration and maximum speed. In early tests, results obtained with a Wimperis accelerometer suggested that the air resistance constant for a tourer car of 1923 was 0.002. A fellow-graduate, Dr. J. N. H. Tait, had also come to the same conclusion with a different make of car. As the years passed, lower values had to be assumed if the resulting calculations were to agree with the known results of performance tests, and values of 0.0015-0.0017 are now in use. There came a time, however, when it was necessary to know the air resistance with more certainty, because of the disappointing results of chassis road tests using a simple box-type test body attached to the chassis for protection of the driver. It was eventually found that this body had 66 per cent greater resistance than a much larger saloon body.

To carry out the tests, the first thought had been to obtain a Wimperis accelerometer, but that proved impossible, and the idea of making an accurate yet sensitive accelerometer rig was conceived. It seemed that a liquid type using an inclined tube similar to an inclined-tube manometer might work, using a tap to adjust damping and to cut off when not in use, this being necessary to retain the liquid in the device when the brakes were applied. Fig. 33 shows the scheme, and the position of the rig in the vehicle. This worked splendidly first time, although some later tests using the more elaborate instrument shown in Fig. 34, have not all been so satisfactory. The scale is arranged to show the low retardation of only 300 lb. per ton extended over a length of some 18 inches. This was accurately calibrated with reference to the levelling block, using an inclinometer reading to 1 minute of arc.

The liquid position against the scale reads zero when the block is level. The whole instrument is then levelled in position in the car with driver and observers seated and with the car standing on a level floor. The total vehicle and passenger weight is taken, and the resistances to motion are measured in pound force per ton weight.

Speed measurement

An accurate measure of instantaneous speed is also needed. A fifth wheel which has been in use since 1931 drives a Hasler tachometer and is shown in Fig. 35. Although the vehicle is decelerating, the mean r.p.m. over the 2 or 3 seconds of test will be recorded, and this is convertible to true m.p.h. The deceleration is read at the instant coinciding with half the time taken to record the speed, and arranged by the driver to occur when the vehicle passes over the spot on the road which is known to be level. The tap is used to "lock" the liquid somewhere near the expected reading, so that when opened a few seconds before taking the reading, lag is reduced to a minimum.

On the road the dead level surface is found with the aid of the instrument and the place is marked. Then simultaneous readings of deceleration (in neutral gear) and m.p.h. (as described above) are arranged. There is a little technique in taking these readings. It is essential that any wind blows substantially in the line of the road so that this factor is added or

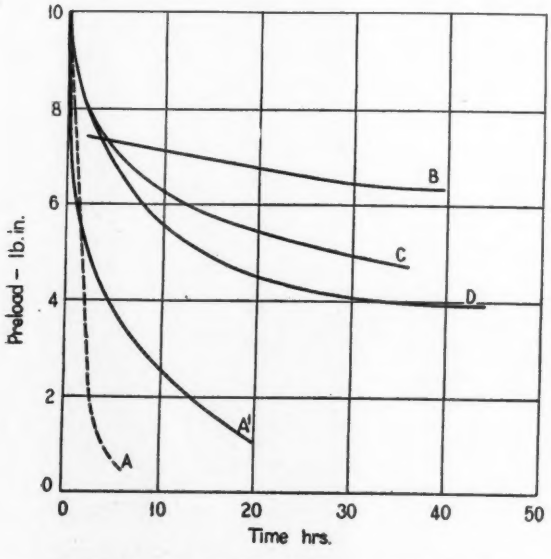


Fig. 31. Results of rig test on rear axle pinion bearing.

With oil filtration; running speed 3,000 r.p.m.; average oil temperature 60–70 deg C; A is with same oil as A¹, but without using filtration system.

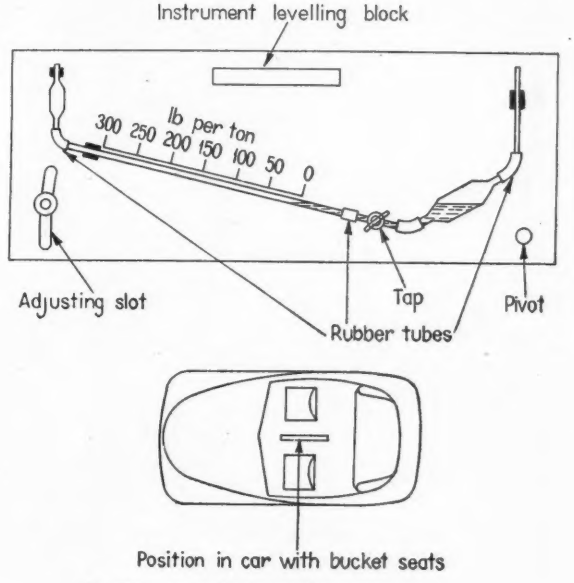


Fig. 33. Inclined-tube accelerometer rig.

Made from laboratory glassware. Instrument set in car with driver and observer seated; quantity of fluid adjusted to read zero when spirit level shows instrument to be horizontal.

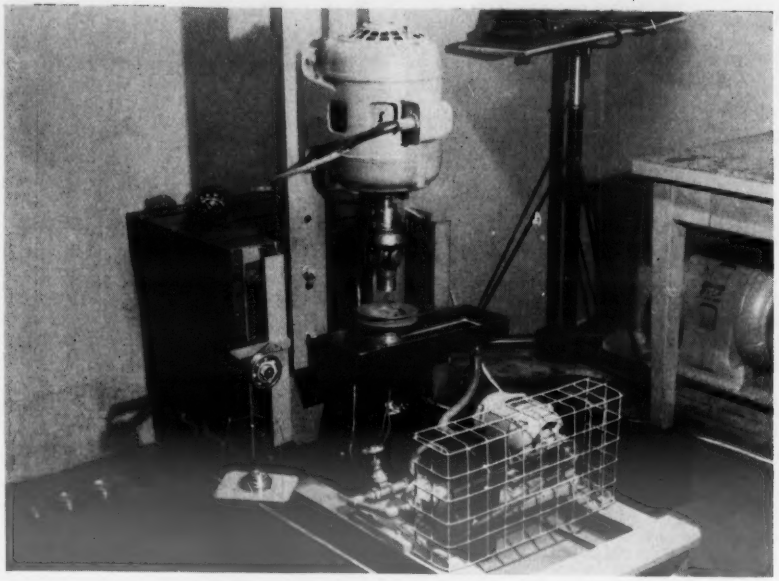


Fig. 30. Taper roller-bearing wear rig.

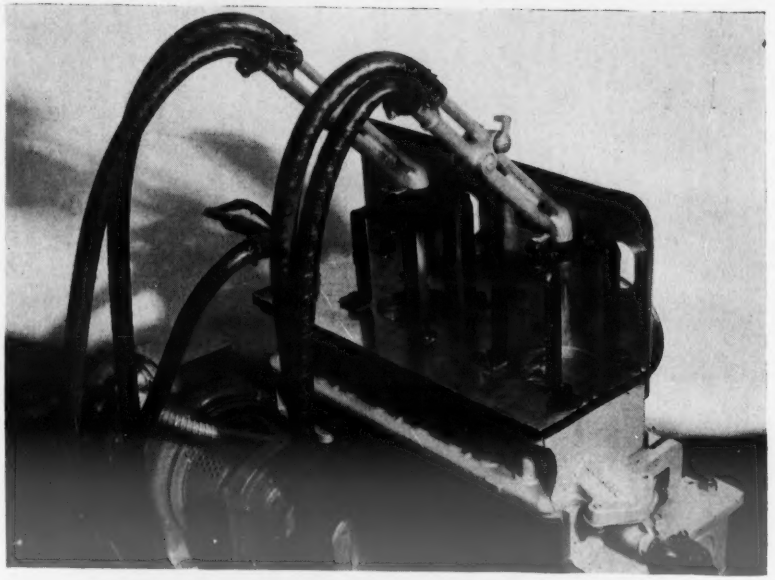


Fig. 32. Rig arranged to run hot exhaust valves at clatter speeds.

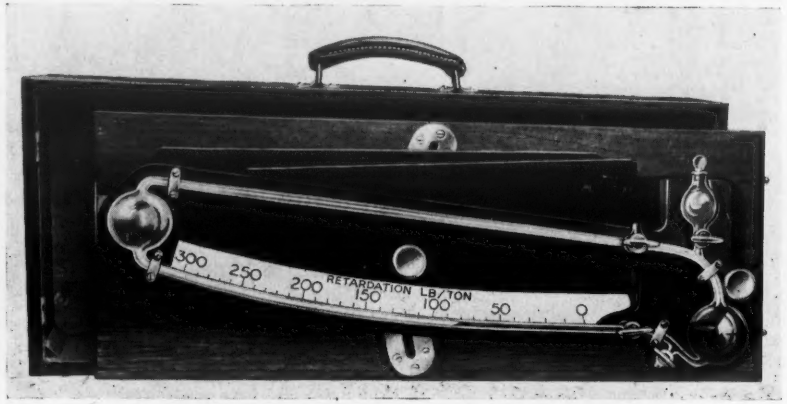


Fig. 34. Curved tube accelerometer developed from original rig.

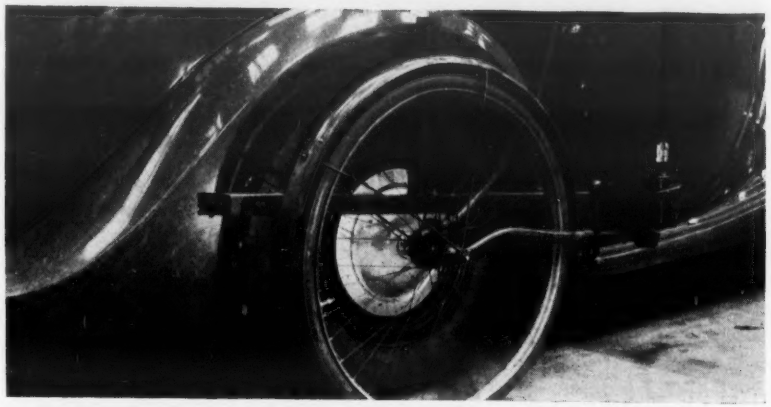


Fig. 35. Fifth wheel.

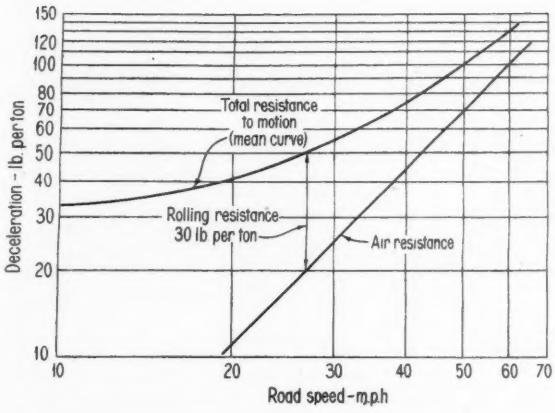

Deceleration - lb. per ton
Road speed - m.p.h
Total resistance to motion (mean curve)
Rolling resistance 30 lb per ton
Air resistance

Fig. 36. Accelerometer test results for 1946, 14 h.p. saloon.
NOTE.—Square law indicated by straight line at 45 deg to base.

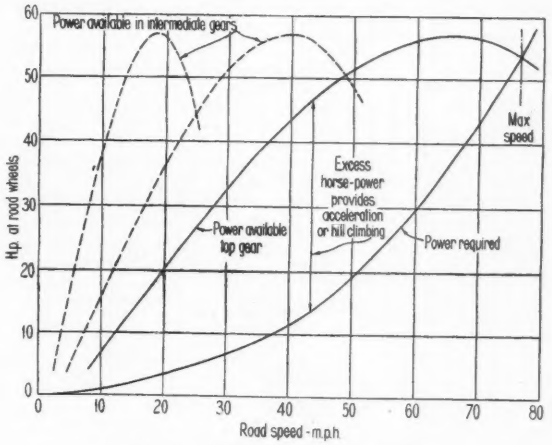

Power available in intermediate gears
H.p. at road wheels
Road speed - m.p.h
Max speed
Excess horse-power provides acceleration or hill climbing
Power available top gear
Power required

Fig. 37. Power-performance curves.

subtracted when the vehicle is tested in both directions on the road. Preferably, the wind should be slight, although it is unlikely to be entirely absent. The car is allowed to decelerate "in neutral" from high speed, so that the car passes over the marked level portion of the road at about the speed at which the resistances are to be determined. These speeds will be 1, 20, 30, 40, 50 and 60 m.p.h. approximately.

At the very low speed of approximately 1 m.p.h. over the level stretch, the rolling resistance is measured. This is assumed constant at the other speeds tested, and subtracted in order to determine the resistance, both ways of the road, which may be assumed to be due to air resistance alone. On many occasions it has been found that plotting the mean of the results, for runs in both directions, shows them to lie on a square law curve, in accordance with the air resistance law.

A convenient way to plot the results is to use logarithmic paper as shown in Fig. 36. In this manner air resistance varying as the square of the speed will then be a straight line at 45 deg.

It will be appreciated that the force of retardation is measured in terms of pound force per ton weight. This weight is that of the vehicle and its load at the time of test and has added to it the rotary inertia effective as added weight. As the gearbox is in neutral, only the road wheels effectively contribute to rotary inertia. Thus the retardation multiplied by the weight gives the total retarding force.

To evaluate the air resistance constant, the frontal area of the vehicle is measured with driver and one observer—the "two-up" condition. The space between the wheels below ground clearance height is not included in the area.

In order that this particular subject may be read in greater detail by those interested, Appendices I and II deal with some practical examples.

In the past the author has obtained square law results by the method of driving at various constant speeds on the level road on a calm day, with provision for recording the angle of carburettor throttle opening. Then, by removing the engine and testing on the bench, the power required at the flywheel for various speeds is determined.

Alternatively, vacuum-gauge readings can be taken and reproduced on the bench, although this method appears to be less accurate. At the present time all these methods are being used simultaneously on a given car, including chassis dynamometer tests, to provide cross-checks and an accurate investigation of this problem.

Conclusion

Although this paper has dealt with several test rigs and has indicated that these are only a cross-section of the large number of rigs operated over a period of two or three years, there is no suggestion that rig testing is more than an aid to the general testing of motor vehicles.

No indoor test can replace the testing of road vehicles on the road or tractors on the farm. Although many test rigs are used, the prime test is on the road.

However, with care in devising the rigs, much useful and quickly obtained results can be acquired in relation to particular problems. All rigs will not be successful, but the great majority should be so.

Test rigs should be used only as an aid, in the same way that mathematical reasoning is an aid to mechanical engineering.

Acknowledgements. The author would like to acknowledge the valuable assistance given to him by his assistants in the contrivance and operation of these test rigs. He is also greatly indebted to the directors of the Standard Motor Company, Ltd., for permission to publish the illustrations and descriptions given in this paper.

Appendix I

VEHICLE PERFORMANCE CALCULATIONS AND FACTORS OF IGNORANCE

This method developed and adjusted in factors over the years has proved remarkably accurate.

The total resistance to vehicle motion on a level road and on a calm day is assumed to be composed of a constant rolling or road resistance plus an air resistance varying as the square of the speed. In terms of horse-power, the constant resistance requires a power varying directly as the speed, and the air resistance a power varying as the cube of the speed. The author has preferred to use horse-power rather than torque in these calculations, because torque is not energy, and only becomes capable of producing motion when allied to a second factor, speed. The product then becomes horse-power.

Table 2. Efficiency Factors

Engine r.p.m.	Efficiency, per cent
1,000	95·2
2,000	94·1
3,000	92·7
4,000	90·9
5,000	88·4

Thus the total horse-power required at the road wheels to equal the resistance to motion of the vehicles on a level road on a calm day is given by:

$$P = \frac{WV}{10} + \frac{xAV^3}{375} \quad .. \quad .. \quad (1)$$

where W is the weight in tons and does not include any allowance for rotary inertia. The rolling resistance is assumed to be 37.5 lb. per ton, V is the road speed in m.p.h., A is the frontal area of the vehicle in sq. ft., and x is the air resistance constant. Values of P are calculated for various road speeds and plotted as shown in Fig. 37, thus giving the curve of power required.

On the same graph is plotted the power estimated to be available at the road wheels when the engine is at full throttle. This is obtained by reference to the engine bench-test results, tested with carburettor road settings and air cleaner, and with fan running, but using the test-house exhaust system. An efficiency factor is then applied to these power values and is to some extent a factor of ignorance, but which for practical purposes serves the purpose of the calculations.

These efficiency factors, which allow for exhaust silencer losses and transmission losses, are given in Table 2 for a range of engine speeds.

It is evident that the power-availability curve can be plotted in relation to car speed for every gear ratio provided in the transmission. There are then sufficient data to estimate the performance with quite remarkable accuracy. These calculations are helpful in determining the most suitable gear ratio long before a prototype vehicle is available.

The instantaneous acceleration of the vehicle is given by the expression

$$a = \frac{5{\cdot}39P_1}{VW_1} \quad .. \quad .. \quad .. \quad (2)$$

where a is the acceleration in ft. per sec. per sec., P_1 is the excess horse-power available (see Fig. 37), V is the instantaneous speed, and W_1 is the weight in tons, including allowance for the effect of rotary inertia.

If the periods over which acceleration is

AUTOMOBILE TEST RIGS (Continued)

Table 3. Moments of Inertia of Rotating Parts

Parts	Polar moment of inertia, lb-ft² units
Crankshaft, etc. ..	0·74
Flywheel	3·22
Clutch	1·30
Total	5·26
Four road wheels, rolling radius 1·095ft..	99·5

Top-gear ratio, 4·625; first-gear ratio, 16·4.

calculated are chosen at intervals of 5 m.p.h. and P_1 is the excess power at the mean speed of these intervals, the time required to accelerate, that is, to increase the speed by he interval of 5 m.p.h. is :—

$$t = \frac{7{\cdot}33}{a} \text{ seconds} \quad .. \quad .. \quad (3)$$

For example, to calculate the time for acceleration from a speed of 25 m.p.h. to a speed of 30 m.p.h., measure P_1 at the mean speed of 27·5 m.p.h. to calculate "a" and so obtain "t".

The use of these formulæ and a tabular method of writing the results on the graph is very convenient.

A Note on Rotary Inertia. The only items which need be allowed for are the moments of inertia of engine crankshaft and connecting-rod big ends, engine flywheel, and the vehicle road wheels. These values are added together taking account of gear ratio. Then an equivalent dead weight w is deduced, such that when added to the vehicle weight, a corrected weight is obtained which allows for the effects of rotating masses, which store energy in rotation in addition to the energy stored due to forward speed of the whole mass of the vehicle.

Example. Consider typical values for a 16 h.p. saloon motor car. Polar moments of inertia for the rotating parts are given in Table 3.

Equivalent added weight effective at rolling radius of road wheels, is, in top gear :—

$$\frac{5{\cdot}26 \times 4{\cdot}625^2 + 99{\cdot}5}{1{\cdot}095^2} = 177 \text{ lb.}$$

and in first gear :—

$$\frac{5{\cdot}26 \times 16{\cdot}4^2 + 99{\cdot}5}{1{\cdot}095^2} = 1{,}260 \text{ lb.}$$

As the vehicle weight with "two-up" is 1·33 tons, the added rotary inertia effect is 6 per cent in top gear, and 42 per cent in first gear. If the first-gear ratio were 20, then this value would be 74 per cent.

Appendix II

SOME EARLY RESULTS USING THE INCLINED TUBE ACCELEROMETER.

The following results are from the first two tests made with the inclined-tube accelerometer.

(*a*) *Fourteen horse-power saloon.* The results measured in both directions of the road in calm weather conditions, were not widely different. The mean values of the results lay very close to the square law, after deduction of the slow-speed rolling resistance, which was assumed constant at all speeds within the speed range of the car.

The curves are shown in Fig. 36.

Analysing the results for a speed of 50 m.p.h. :—

Weight, including observers, as tested	3,155 lb.
Allowance for effect of rotary inertia of road wheels	80lb.
Effective total ..	3,235lb. = 1.44 tons

From Fig. 36, the retardation due to air resistance is 69lb. per ton. Thus, the total air resistance, F, is 99·5lb. The projected frontal area of the car is 23·2 sq. ft.

Now, $x A V^2 = F$

From which, $x \times 23{\cdot}2 \times 50^2 = 99.5$

Thus, the air resistance constant

$x = 0{\cdot}00172$.

This compares with the value of 0·0016 used in calculations for this car. Further work is required to satisfy the author of the accuracy of these measured results; there is a discrepancy of 7 per cent between the "guessed" value of 0·0016 and the apparent measured value of 0·00172.

(*b*) *Chassis with Box Body.* Here again, the test results for one direction of the road lay within 6 per cent of those for the other direction and were even closer at the highest speeds. The tested mean figures, after subtracting the 45 lb. per ton constant for rolling resistance for this vehicle measured at very low speed, are as shown in Table 4. In this case, the projected frontal area was 21·75 sq. ft., and the effective weight as tested was 3,260 lb. Thus, calculating as in (*a*) above, the air resistance constant $x = 0{\cdot}00278$.

Table 4. Retardation Test Results of Air Resistance for Chassis with Box Body

Road speed, m.p.h.	Retardation, lb per ton	
	As tested	Square law comparison
20	16·5	16·7
30	36·5	37·5
40	66	66·6
50	107	104

If this factor is used in the performance calculation as described in Appendix I and comparison is made with the actual performance of the vehicle when road tested, the results shown in Table 5 are obtained. The acceleration values are obtained, not by snap opening of throttle at the lower speed of the tested range, but by starting at a lower speed and by timing while "passing through" the speeds.

These results confirm the high air resistance of this temporary box-shaped body, and stress the effect of extreme shapes even at speeds appreciably below maximum. For instance, the powers required to maintain a uniform speed of 60 m.p.h. on a level road are shown in Table 6.

These two vehicles have equal frontal area of 23 sq. ft., equal weight of 3,000 lb. and equal rolling resistances of 30 lb. per ton.

Appendix III

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RICARDO, H. R., 1931 "The High-speed Internal Combustion Engine" (Blackie and Son, Ltd.).

Table 5. Calculated and Tested Performance Data for Chassis with Box Body

Conditions	Calculated	Tested
Maximum speed	67·6 m.p.h.	66·9 m.p.h.
Acceleration :		
10–30 m.p.h.	10·1 seconds	9·6 seconds
30–50 „	11·8 „	12·1 „

Table 6. Powers required to Maintain Uniform Speed of 60 m.p.h. on Level Road

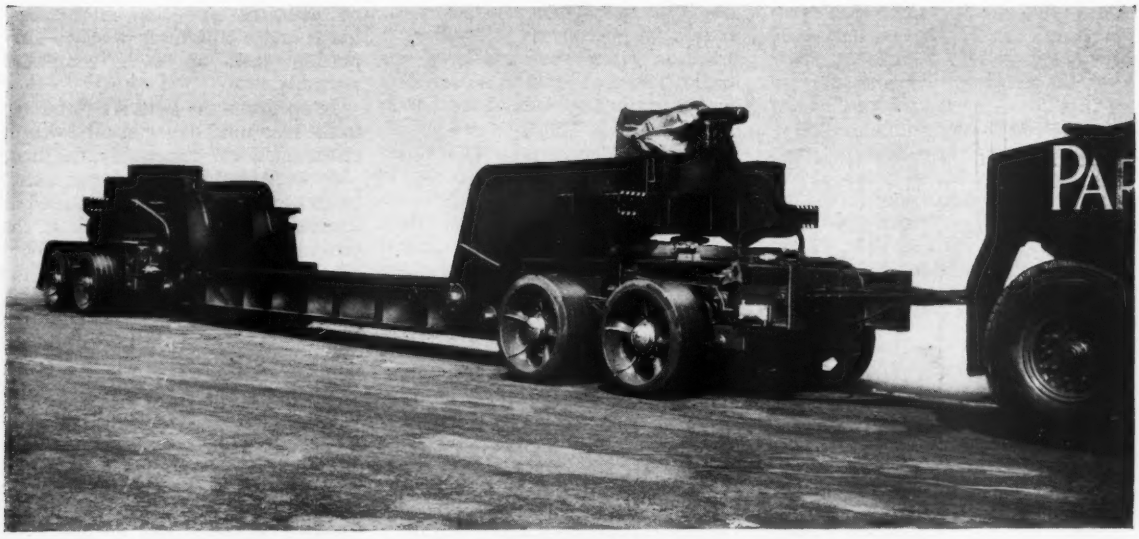
Item	14 h.p. saloon	Chassis with box body
For rolling resistance ..	6·4 h.p.	6·4 h.p.
For air resistance	22·8 „	36·8 „
Total	29·2 „	43·2 „

Gasoline Manufacture

FOUR films in the Refinery Processes series have recently been completed by the Shell Petroleum Co., Ltd., and although they are primarily intended for staff instruction, the needs of schools, universities and technical colleges were also kept in view. The first film entitled Octane Number, explains compression ratio, its relation to power output, and the way in which knocking limits this power, and concludes with an exposition of octane number rating. Cracking is dealt with in the second film, which is concerned with the refinery processes by which heavy oils are converted into gasoline.

The third and fourth films, namely Components for Aviation Gasoline and Aviation Gasoline—Blending, concern the manufacture of aviation gasoline where the highest quality is demanded. The chemical processes shown are more complex than in cracking, and the physical mixing or blending of various kinds of gasoline is explained. Much use is made in the films of animated diagrams and molecular models, and the subject is presented in an admirably clear manner. (1906)

The Mond Nickel Co., Ltd., Sunderland House, Curzon Street, London, W.1, have recently issued a booklet dealing with the composition and properties, machining, welding, brazing, soldering and founding of nickel-aluminium bronze.



The 100-ton sixteen-wheeled Dyson trailer.

THE DYSON TRAILER

A 100-ton 16-wheeled Design for Special Transport

THE transport by road of heavy and bulky machinery in this country is made particularly difficult by the limited headroom provided by many of the old railway bridges crossing the public roads. In some instances an improvement has been made by making a dip in the roadway under the bridge, but this may be of little value when the vehicle is so long as to bridge the lowered portion.

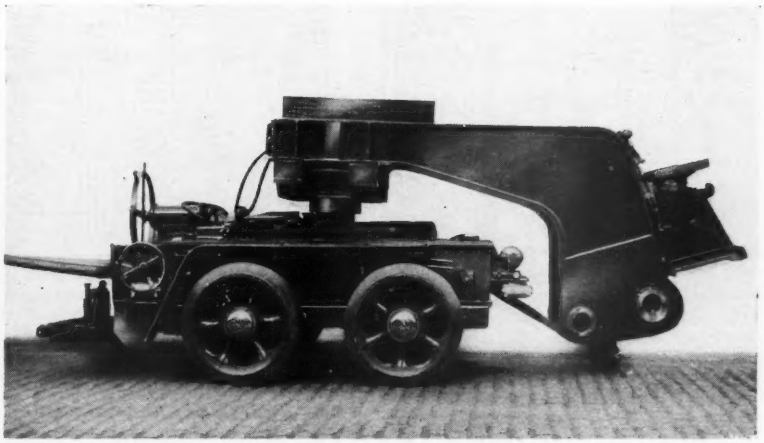
While every effort must be made to keep the frame of a load-carrying vehicle as low as possible, this in itself causes trouble on some overbridges which have steep approaches and a humped back. Raising and lowering of the frame during a journey is frequently necessary.

In some cases, such as the Scammell 100-tonner described in *The Automobile Engineer* for March, 1930, one end only of the frame is raised and lowered, hydraulic jack-struts being used in the connection of a hinged swan-neck member to the frame girders. A very large movement (about 2 feet) was provided and could be made in a few minutes without uncoupling any parts. The front end of the frame could be dropped until it nearly touched the ground and the vehicle could be driven in this form if necessary.

At the same time the large movement permitted the load to be relieved from the rear bogie by placing packing under the girders towards the rear and lowering the front end, the rear bogie being then run away and the frame finally lowered till the girders rested on the ground after raising again and removing the packing.

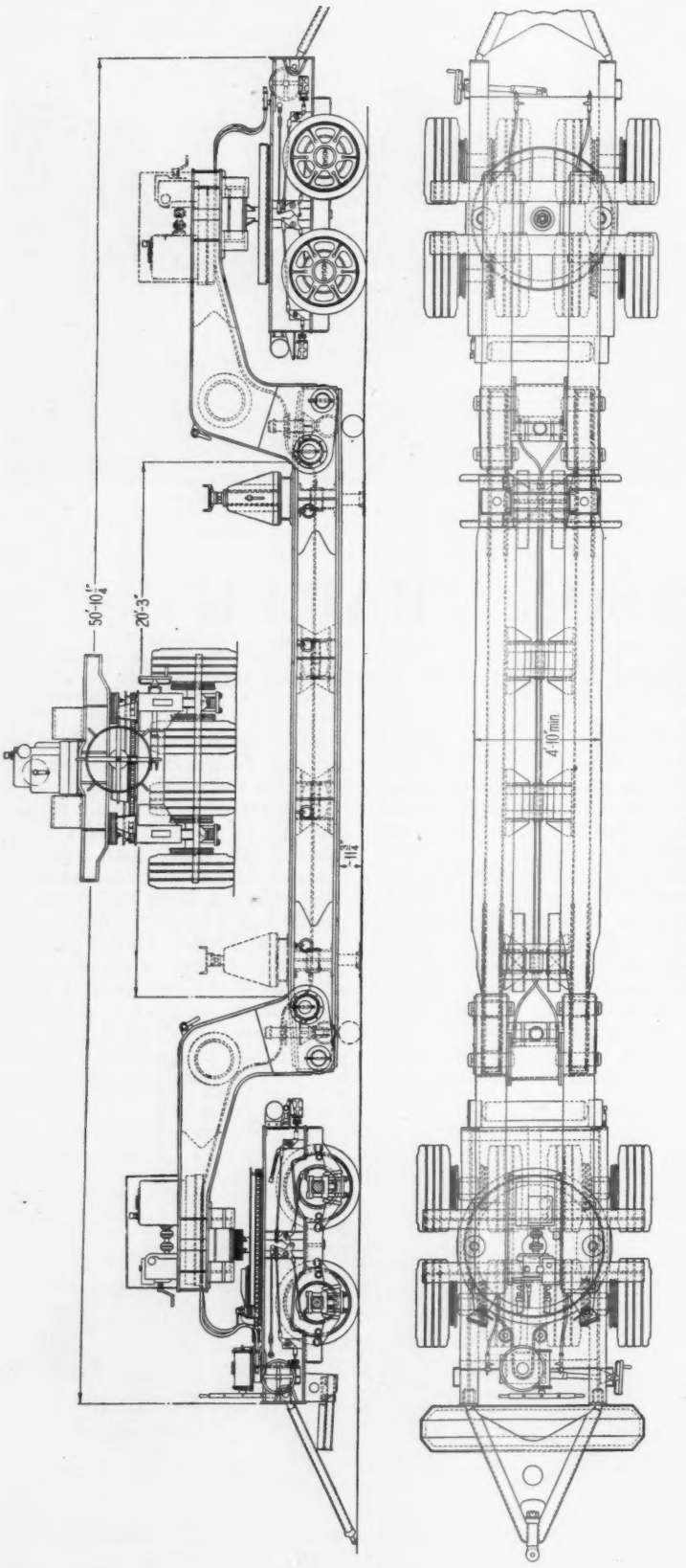
The Scammell arrangement necessitated a fairly long base to the swan-neck to keep the load on the jack-struts to a reasonable figure. In the case of the Dyson trailer it was considered necessary to keep the trailer as short as possible so that jacking struts could not be employed. Instead each of the two box-section welded frame side members are joined by four detachable cross members, the end ones of which carry two hydraulic jacks each, making four in all.

The swan-necks are in pairs and are connected to the side members by 6in diameter pin-joints. Each inner pin passes through eccentric bushes in the swan-neck, and these bushes can be rotated half a turn, to either top or bottom dead-centre. This gives an angular displacement of the swan-neck resulting in 5¼in difference in frame height from the road. The front bogie bolster carries a 3 H.P. air-cooled petrol engine driving a pump supplying, through quickly detachable con-



Rear bogie and swan-neck.

THE DYSON TRAILER (*Continued*)



General arrangement of Dyson 100-ton trailer.

nections, the four jacks on the main frame and a subsidiary wheeled supporting jack on each swan-neck assembly.

In operation the jacks are run down to the road until they carry the weight of the main frame assembly, the inner pins are withdrawn and the eccentric bushes rotated and the frame raised or lowered, as the case may be, until the pins can be re-inserted. Alternatively all pins can be withdrawn and the frame lowered right onto the road, when the bogies, together with the swan-necks carried by the subsidiary jacking rollers, can be wheeled away.

As shown in the drawing the main-frame side members are at 3ft 4in centres, but additional cross members of various lengths are provided so that the frame and swan-necks can be assembled at centres up to 7ft 11in. The machine has been built to the special requirements of Sir Lindsay Parkinson and Co., Ltd., for the transport of drag-line excavators of various sizes. The smaller ones are carried with their caterpillar tracks resting on the top flanges of the side members, but, when carrying the larger ones the frame members are set closer with the tracks hanging down on each side, thus considerably reducing the overall height of the load.

Each bogie consists of a fabricated box carrying four axles, oscillating on leaf springs having Woodhead trunnion mountings at their ends. The wheels run on Timken roller bearings and are 771 mm in diameter, fitted with single solid tyres 15in wide and rated at 8 tons each.

While the main frame is adjustable in width, the bolsters have an overall width of 8ft 6in and have drilled holes for the bolting of the swan-necks at the various widths. Each bolster transmits the load to its bogie by a central ball joint, being additionally steadied by two pads ball-jointed to vertical plungers. These plungers are fixed in the case of the forward bogie, but on the rear bogie they have rubber disc cushions to allow slight rocking of the bolster to take up road inequalities.

The forward bogie has the usual turntable and pressed-steel drawbar. The rear bogie has a similar turntable, but it is provided with locking pins securing it in the straight ahead position for normal running. When negotiating sharp corners the turntable can be steered by hand. A large wheel with pilot handles is secured to a horizontal wormshaft, meshing with a wormwheel on a vertical sprocket shaft from which a heavy Renold roller chain runs, round two idlers, to the turntable ring. A drawbar can also be fitted to the rear end if desired, the

turntable locking pins being then withdrawn permitting the drawbar to take charge of the bogie without stressing the hand steering mechanism.

2LS Girling brakes, 16½in diameter by 5in wide, are fitted to all wheels, and are operated by linkwork from four Clayton-Dewandre air pressure cylinders on each bogie and also by a handwheel and compensating mechanism. The compressed-air brakes are piped up to the tractor, which is a Scammell rigid six-wheeler, the two-line system being employed. A platform is provided at the rear of the back bogie on which the rear steersman can ride; he also has control of the hand-brake on the rear bogie.

The overall length of the complete machine is 50ft 10½in, the wheelbase of each bogie is 4ft 4in and the bogie centres are 38ft 11½in. The clear space available on the top flanges of the frame side members between the swan-necks is 20ft 3in, part of this space being normally occupied by a detachable and adjustable dolly which supports the cab of the excavator when travelling. The extreme width over the wheels is 8ft 11in and over the bolsters 8ft 6in.

In the lower position of the frame the unladen ground clearance is 11¾in and the frame height 2ft 10½in. These figures are increased by 5½in when the eccentrics in the swan-necks are reversed, and they are all reduced by about 1½in by the deflections of springs and frame under full load.

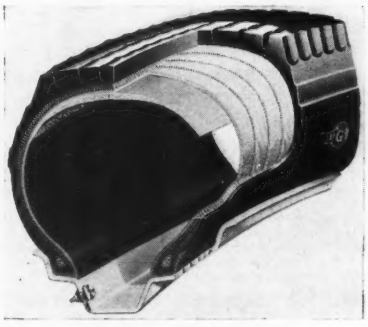
A NEW TYRE

A Tubeless, Puncture-Sealing Tyre by the B.F. Goodrich Co.

FOR more than ten years the B.F. Goodrich Company of Akron, Ohio, has had under development a tubeless tyre for automobiles. During the war period intensive research was conducted on special tubeless tyres for military vehicles and also on bullet-sealing compounds for tyres and fuel tanks. The culmination of these efforts is the production of a puncture-sealing, tubeless tyre for the private car.

It is reported that more than a million miles of operation under most arduous conditions have been accumulated by these tyres fitted to test vehicles, police cars and taxicabs. At present, they are available to the public in the State of Michigan but by the end of this year they should be distributed throughout America.

When fitted on a wheel the external appearance in no way differs from that of a conventional tyre. The casing and tread is of a tough, wear-resistant, "cold" rubber and in addition to the usual fabric plies there is a thin inner ply providing an effective air seal. Differing from some other designs, the sealing of the beads is effected directly on the rim flange and not by a separate internal device. It follows that a tyre of this type cannot be used on a



The Goodrich tubeless tyre.

divided rim or a rim having a detachable flange. The outer face of the bead is formed with a series of concentric ridges or lips which when forced by the tyre pressure against the flange, constitute a multi-channel seal. A normal valve is mounted in a sealing bushing directly in the well of the rim.

Inside the crown and shoulders of the casing is a layer of a puncture-sealing material. If a sharp object pierces the tyre it is tightly gripped by this viscid compound which, when the object is withdrawn, is pulled into the hole to effect a permanent seal against loss of air.

Compared with a conventional tyre and tube assembly, the tubeless tyre is stated to retain air better, give a more comfortable ride, to run cooler and thereby reduce the risk of blow-out, and to eliminate the hazards of pinching or chafing. It is essential, of course, that the wheel rim be in sound and clean condition. The flanges on which the seal is made must be cleared of rust or oxidized rubber with steel wool or a rotary scratch brush. Any dents or distortions in the flanges must be hammered out and the butt weld examined and, if necessary, filed to eliminate either depressions or high spots. Any loose rivets should be clinched or renewed but must not be welded or brazed.

The tubeless tyre is fitted with a temporary spacer during transport and storage in order to prevent flattening or bead distortion which could make initial inflation difficult. Ordinary procedure is followed for mounting or removing the tyre, with special care not to damage the bead-sealing lips. Tyre levers must be free from burrs, and should not be used to "break" the bead from the rim when demounting.

Automatic Tuning Mechanism

SALES, design and manufacturing rights for a new "Autoselector" automatic tuning and selecting mechanism have been granted to The Equipment Division of Mullard Electronic Products Division, Century House, Shaftesbury Avenue, London, W.C.2, by Philips Telecommunications Industries. This device is used for turning shafts repeatedly to pre-set positions with speed and accuracy. Automatic electrical remote control is employed. The setting accuracy is such that the positions selected never differ by more than plus or minus 0.050 deg from those which have been pre-set in the machine. This degree of accuracy does not vary with different operators, speed of operation or other variation in conditions. It would appear that this device may have applications where extremely accurate machine tool operations have to be repeated many times. (1898)

There has been no change in the position with regard to the withdrawal of overtime working by a section of the printing industry. A slight reduction in the number of pages in *The Automobile Engineer* continues to be unavoidable. All journals printed in London are similarly affected, to a greater or lesser extent, but journals printed in the provinces are not affected.

Silicone Rubber

THE Dunlop Rubber Co., Ltd., have set up a new unit at their works in Cambridge Street, Manchester, for the manufacture of Silicone rubber products. Silicone rubbers are produced in various grades of hardness and mechanical properties, but they all have the characteristic of retaining rubberlike qualities under conditions of extreme heat and cold. It is claimed that they have an economical life throughout the temperature range of minus 70 deg F to plus 400 deg F. Among the articles being made are gaskets, seals and packings, sealing rings, vibration dampers, hot-air ducts and insulating parts for a variety of purposes. (1908)

RUBBER SUSPENSION

A Motor Cycle Application by the Dunlop Rubber Company

A SUSPENSION system for motor cycles using rubber stressed in compression, has been developed by Dunlop Special Products, Ltd., Fort Dunlop, Erdington, Birmingham, 24, and suspension units suitable for front and rear wheel springing are now available. Regarding the front suspension, this closely resembles externally the telescopic fork in general use. Reference to the illustration will show that each leg of the fork comprises three main sub-assemblies, namely inner and outer telescopic tubes and a "cartridge" carrying the rubber springs. This cartridge is carried inside the inner tube and consists of a central control rod, formed with a flat on one side, over which the rubber is assembled.

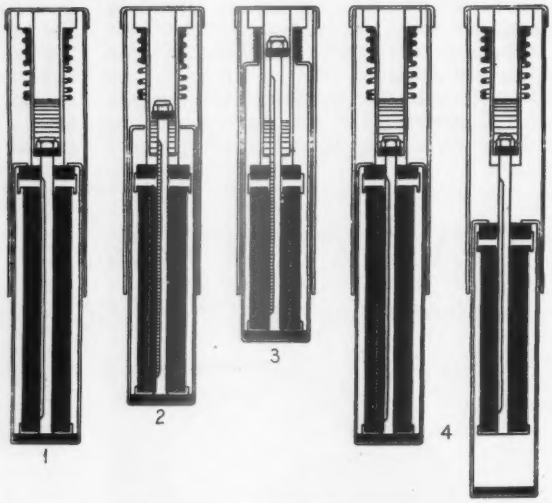


Diagram of cycle of movement.

1. Machine unladen. 2. Riding position with main rubber spring depressed and fluid drawn down to the rubber element. 3. Maximum bump position. Main rubber spring depressed to full extent, overload spring compressed and rubber limit stop in operation. 4. Strut reflexes through 2 and 1 to maximum recoil. Main rubber spring compressed in reverse direction with fluid returned to reservoir.

There is a difference of 0·030in between the bore of the rubber and the rod so that in the free position the rubber grips the rod. Since damping depends to a large extent upon the grip of the rubber, it is essential that the bore of the rubber should be perfectly true. For this reason accurate control of bore in manufacture is necessary.

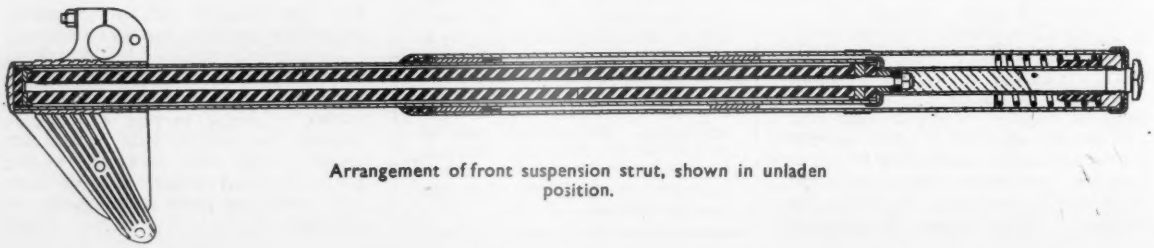
Carried within the outer telescopic tube is a top post or plunger, against the bottom end of which the cartridge is compressed on bump, while the inner tube passes over the post. On rebound the central control rod, the upper end of which is retained by a nut within the top post, and which is provided with a piston washer at its lower end, compresses the rubber against a screwed cap at the top of the inner tube. Two bearings carried in the outer tube provide the necessary guidance for the inner tube, which is chromium plated on the bearing surface. The bore of the top post forms a fluid reservoir and in the case of a natural rubber cartridge, is filled with a castor based oil. A fluid film is maintained between the rubber spring and the control rod and this has an important bearing upon the characteristics and sensitivity of the suspension.

Referring to the diagram showing the complete cycle of movement, position 1 represents the condition with the machine unladen. In the riding position, shown at 2, the additional weight has compressed the suspension to its static laden position and the upward movement of the inner tube and control rod has opened a rubber valve retained by the nut at the top of the control rod. The control rod has moved further into the fluid reservoir so that the flat machined on its side provides a passage through which fluid can pass to the bore of the rubber element. This action is assisted by the fact that under compression the bore of the rubber expands. Position 3 shows the maximum bump position in which the rubber element is compressed to its fullest extent and the screwed cap at the top of the inner tube has not only compressed the over-load coil spring, but has made contact with a rubber limit stop carried at the top end of the top post.

As the strut returns to its normal position fluid is forced back into the reservoir and the valve closes to prevent loss of fluid during extension of the strut on rebound. Position 4 shows maximum rebound with the main rubber element fully compressed in the reverse direction and with the fluid sealed off in the reservoir.

In connection with the over-load coil spring and rubber limit stop, it should be noted that the spring is a tight fit on the limit stop and when compressed it expands outwards, releasing its grip on the stop before the stop itself comes into operation. In the reverse direction the relative movement between the coil spring and the rubber stop has a damping effect which, it is stated, ensures silence in operation. It is also claimed that since the valves in the fluid reservoir are of a simple type they require no attention in service and that the only maintenance necessary is occasional topping up of the fluid reservoir in the top post. Damping during rebound is provided by hysteresis of the rubber and by its increasing grip on the control rod as it passes the normal static laden position. Typical suspension ranges for a lightweight machine are $4\frac{1}{2}$in bump and $2\frac{5}{8}$in rebound.

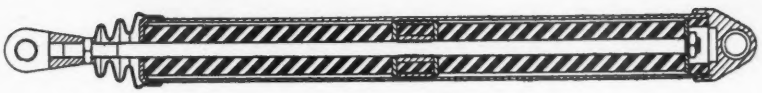
In its application to the rear sus-



Arrangement of front suspension strut, shown in unladen position.

pension of a motor cycle, the Dunlop strut embodying rubber cartridges may be used in a triangulated arrangement whereby the rear wheel is carried in a fork pivoted to the main frame adjacent to the gearbox sprocket. The two struts, one each side of the machine, are each attached at their inner ends to the frame of the machine and at their outer ends to the swinging fork at a point adjacent to the rear hub spindle. The struts may be arranged above or below the fork, but in either case a triangulated structure is provided. The strut comprises an outer tube, control rod and two cylindrical rubber springs pre-compressed on to the rod, as in the case of the front suspension. The two springs work in parallel, and since there are two struts in the complete suspension, each spring carries one-quarter of the load.

On bump, the control rod is subjected to tension and compresses the rubber springs, the inner one by means of a free piston washer retained by a nut at the end of the rod, against a free piston washer, the movement of which is limited by an abutment midway along the outer tube. The outer spring is compressed by a second free piston washer butting against a sleeve attached to the rod, against a screwed retaining cap at the end of the tube. Ultimately the spring movement is limited by the compression and expansion of the rubber against the bore of the outer tube.



Rear suspension strut in unladen position.

On rebound, the outer rubber spring goes out of action when the no-load position is reached, since its free piston washer rests against the abutment in the outer tube, while the inner spring is compressed through the medium of the adjacent free piston washer, against the piston washer at the inner end of the assembly. Rubber sleeves are fitted at the centre abutment of the control rod and at the inner end of the strut adjacent to the end piston washer, ensuring silence and smoothness of operation by reason of a slight pre-load effect. Typical deflections of the strut for a lightweight machine are 3½in bump and 1½in rebound. It is stated that with the full 3½in deflection of the rear suspension, damping is complete in two oscillations.

Although the first demonstration of the Dunlop suspension has been on motor cycles, the principle evidently has a wide field of application, not only as a suspension system for road vehicles, but also in the form of stabilizers, etc., in association with other mechanisms.

CORRESPONDENCE

Correspondence on subjects of technical interest is invited. The name and address of the writer must be given, though not necessarily for publication. No responsibility is accepted by the Editor for the opinions of correspondents, and the right is reserved to omit any portion of a letter. If a reply by post be desired, a stamped addressed envelope should be enclosed.

CARS OVERSEAS

SIR,—Your article in the February issue of your journal on "The Specialist Car" was read with interest. It could be concluded, however, that the problem of the mass produced low-priced car has been solved as far as British manufacturers are concerned.

While I personally have a great deal of time for English vehicles, and would, in fact, choose an English car in preference to an American product for my personal use, I have found, in controlling a fleet of over 1,000 vehicles, that, from the angle of overall maintenance costs and, in fact, overall costs including petrol and oil, the English machines up to 20 h.p. are not necessarily cheaper to run than an American counterpart of 30 h.p.

At the present time, the British manufacturer has a golden opportunity of offsetting the effects of years of prejudice. This prejudice has been built up because of English vehicles inherently unsuited to Australian conditions which have been condemned by users of American vehicles of much heavier construction and h.p. Especially with today's high output from small engines and rising petrol costs, a vehicle thoroughly designed for Australian conditions could become established and ultimately replace the many functions for which the American car is now considered to be the only suitable vehicle. Such a car in the commercial field (utilities and panel vans) would require the following general specification:—

Engine: 16 h.p., medium compression, with overhead valves (an engine on the pattern of the Jeep motor would be ideal, but overhead valves are necessary for low maintenance costs).

Gearbox: Of constant mesh, non-synchro type, with centre floor gear lever.

Brakes: Of the hydro-mechanical type, with facilities for inspecting linings without dismantling.

Transmission: Of the general dimensions normally supplied for a 25 h.p. vehicle.

Body: To be constructed with a view to easy maintenance; cabin should have exceptional rear vision, and mudguards should be of rubber.

Windscreen wipers: To be vacuum operated from a vacuum pump incorporated in the petrol pump.

Dashboard: To be extremely simple, with indicator lights for oil pressure and temperature. If possible, the dashboard should jack out as a unit to permit quick replacement.

A vehicle to this specification would be very popular in this country.

E. W. CORLESS.

East Ivanhoe,
Australia.

Standard Bronze Bars

SOME time ago the Glacier Metal Co., Ltd., Alperton, Wembley, introduced a standard range of chilled cast phosphor bronze bars in 2B8 specification, machined on the outside diameter.

There is a standard range of 153 sizes of bars machined both on the outside diameter and in the bore now announced in three specifications, namely, chill cast 2B8 phosphor bronze, Glacier LB9 lead-bronze and Glacier LB20 lead bronze.

The bars are all supplied in 13in lengths, cored bars being available with outside diameters ranging from 1in to 6in and solid bars with outside diameters of from ⅝in to 6in. Cuts of only 0·010in on the outside diameter and 0·015in. in the bore are needed for finishing, and accordingly it is claimed that as a result there are considerable savings in metal, tool wear and machining times. (1913)

Wax Sprayer

A NEW wax spraying outfit designed for spraying cars with liquid wax polishes is announced by the Aerograph Co., Ltd., Lower Sydenham, London, S.E.26. The Aerograph De Vilbiss Type AP-659 outfit includes spray gun, air transformer, pressure regulator and a 25ft length of air hose with connections. The equipment delivers a fine uniform spray that dries quickly into an effective wax coating. (1910)

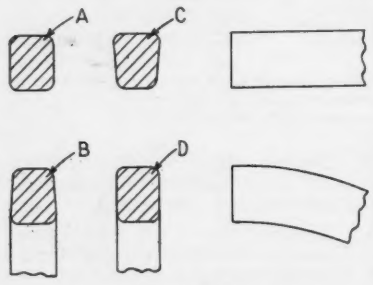
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Production of starter rings

STARTER rings are commonly formed from a length of rectangular steel bar A bent to circular shape and welded into continuity. A disadvantage of this process is that the metal becomes stretched at the radially outer half of the section and compressed at its inner half with the consequence that the resulting ring is of a wedge section, as at B. As the finished ring must be of rectangular section it has to be machined on both side faces and the amount of metal to be removed renders it impracticable to effect by a grinding operation.

The invention proposes the use of a steel bar C rolled to a suitable wedge section so that on bending into a ring it assumes a rectangular section, as at D. It may be possible to cut the teeth on the ring in this condition, but if necessary or desirable the side faces can be finished by surface grinding. *Patent No.* 618298. *W. G. Electrical Welding & Engineering Co., Ltd., and L. Guest.*

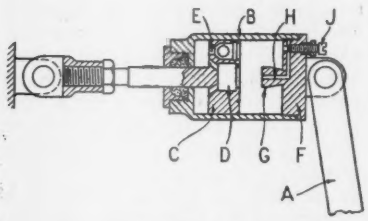


No. 618298

Clutch actuation control

TO obtain smooth engagement of a clutch it has been proposed to use a dashpot device but a controlled movement at uniform speed slows down the operation and may constitute a disadvantage, particularly when travelling up hill. This invention provides a hydraulic dashpot with a suitably varied rate. Linked to an arm A connected to the clutch pedal is a cylinder B slidable over piston C pivotally anchored to the engine casing. The piston is furnished with a chamber D with a reduced diameter orifice on one face and drillings to the other face to put it in open communication with both ends of the cylinder. It also has a plurality of passages E each closed by a non-return ball valve. Cylinder cover F has an axially arranged coned spigot G which, on the engaging movement of the clutch pedal, enters the chamber in the piston. The major diameter of this spigot virtually closes the orifice of the piston chamber and the transfer of fluid is then by way of drilling H, the effective cross-section of which is adjustable by means of screw J.

The sketch shows the device when the clutch is disconnected. As the pedal is released the cylinder moves over the piston, the ball valves close, and the oil moves through the chamber and drillings to the opposite end of the cylinder. This provides the initial rapid movement but



No. 619150

as the spigot enters the chamber orifice the flow is restricted through passage H and the relatively slow, smooth take-up of the drive is effected. As soon as the major diameter of the spigot has entered the chamber the reduced section of the spigot permits a more rapid flow and thus a quick finishing of the stroke.

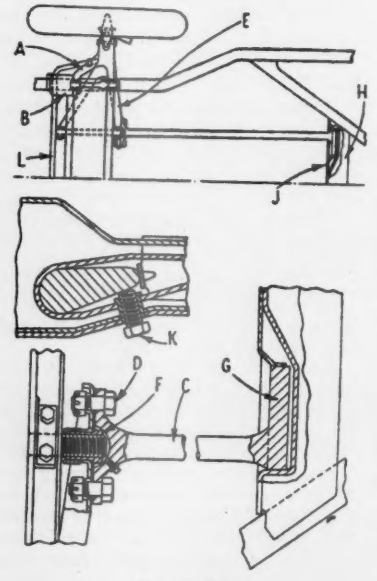
Disengagement of the clutch is, of course, not impeded as the ball valves lift to permit a speedy transfer of the fluid. Regulation of the slow movement is effected by adjustment of the screw J while the setting for the commencement of the slow movement is accomplished by adjusting the position of the piston in relation to its anchorage yoke. *Patent No.* 619150. *C. Prachar. (Czechoslovakia.)*

Torsion spring suspension

THE object of this invention is to avoid the expensive machining operations and close tolerances involved in the use of splined connections for the torsion spring. The wheels are each connected to the vehicle framing by a pair of vertically spaced wishbone levers pivoting on threaded bushings and upper lever A carries the shock absorber B. At its forward end the torsion bar C is upset to form an elongated flange by which it is secured by two substantial bolts D to the lower wishbone lever E in alignment with the threaded bushing F on the pivotal axis. An axial cavity in the face of the flange serves to enclose the end of the pivot shaft and forms a chamber for lubricant introduced by way of a nipple.

To effect an anchorage, the rear end of the torsion bar is upset to form an integral lever G which is received in a socket pressed in the forward flange of transverse frame member H. The semi-circular profile of the lever, co-axial with the pivot axis, is seated in one end of the socket by means of bracket J and permits angular adjustment of the rear end of the bar by means of screw K to vary the riding height of the vehicles.

A transverse sway bar L is pivotally mounted on the chassis frame and linked to the outer ends of the lower wishbone members. *Patent No.* 620014. *Ford Motor Co., Ltd.*



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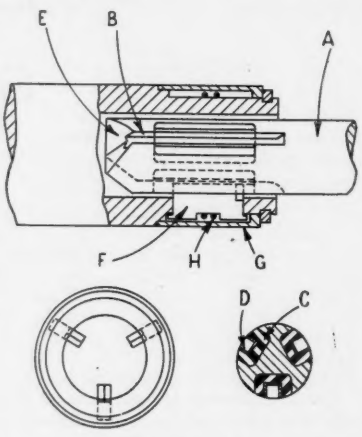

No. 618534

Auxiliary drive coupling

THIS torque-transmitting coupling, particularly intended for the drives of auxiliary equipment, is of the slidable splined type with resilient elements permitting a limited degree of misalignment and possesses features facilitating engagement. Drive shaft A is formed with three grooves B, each furnished with a recess to house a resilient member C carrying a channel-section bearing member D, which forms a continuation of groove B. At the extreme end of the shaft the grooves are flared out to form sloping guide faces E.

The end of the driven shaft is bored to receive the drive shaft and slotted to accommodate three keys F which make driving engagement with the grooves B. Radially inward movement of the keys is limited by projecting lips whilst a sleeve G retained by a spring ring limits radially outward movement. The keys are urged inwardly by a helical spring H. At their leading edges keys are tapered to co-operate with the sloping faces E when the two elements are presented for engagement. *Patent No.* 618534. *Self-Priming Pump & Engineering Co., Ltd., J. A. McKean and A. L. B. Dawson.*